Heating · Piping and Air Conditioning

JULY, 1931

Articles by—

E. H. BARRY

I. C. BAKER

W. H. PIERCE

S. R. LEWIS

M. W. BENJAMIN

W. H. WILSON

Journal Section of A. S. H. V. E.

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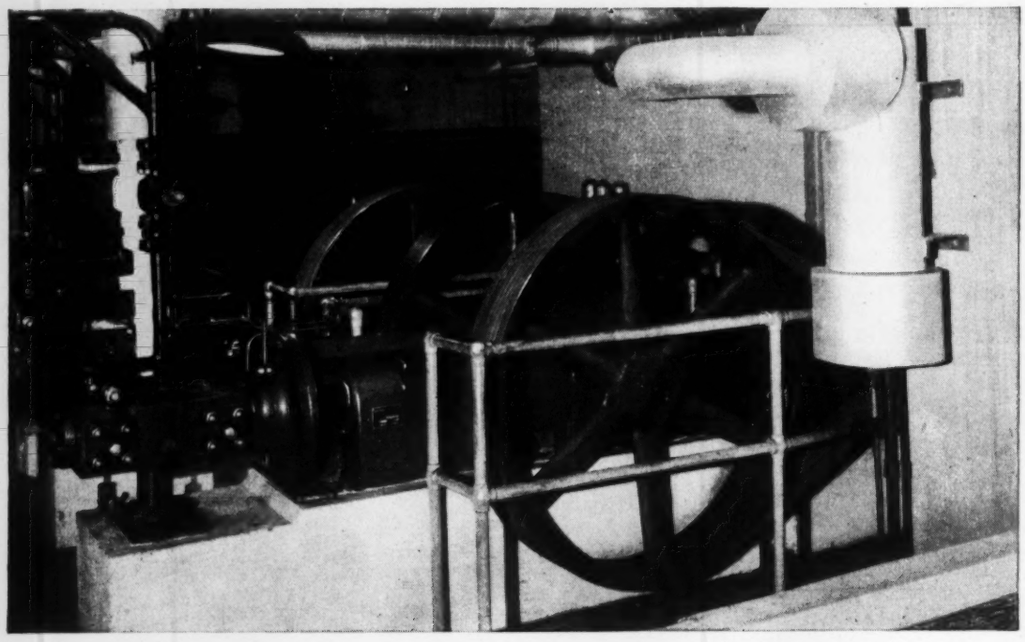

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The EDITOR'S PAGE

Heating · Piping and Air Conditioning

FROM ARTICLES IN THIS ISSUE

"When steam is used in process, it serves as a conveyor of heat. Usually the substance of the steam is not consumed in the manufacturing process. The manufacturing departments require delivery of heat by the power plant at a given rate. . . . This fact must be considered when the design of a high pressure power plant is contemplated."—*E. H. Barry.*

"The increasing popularity of the tall building has introduced a set of outside conditions which often require a chimney to be carried to a height not necessitated by the actual draft requirements. In such instances the diameter may be reduced correspondingly, to the advantage of a lesser first cost and an increase in rentable floor area."—*H. L. Alt.*

"One very important point to remember in piping steam to a liquid soap system (in a laundry) is the installation of a tee with a swing check valve opening to the atmosphere. This is in case pressure goes down in the boiler at night, thus creating a vacuum which will tend to pull liquid soap from the supply tank back to the boiler."—*W. H. Pierce.*

"Local fan-equipped unit coolers supplied with chilled water or which use direct expansion refrigerants . . . are available. One may install a compressor somewhat larger than that required for the ordinary large ice box cooler in the basement, serving from it a unit cooler which may stand on a pedestal in the private office, promoting wonderfully pleasant conditions in a very simple and easily controlled manner. — *Samuel R. Lewis.*

"This equipment was not regularly inspected by the maintenance employes and the faucet was allowed to remain in this condition for some time. It was estimated that 8000 gallons of hot water per month were wasted in this manner."—*W. H. Wilson.*

"Central heating was first undertaken (in Winnipeg) as an economical measure to provide means of utilizing an electrical stand-by plant. Electrically heated boilers . . . provide an outlet for off-peak energy."—*Some Reports on Research.*

FROM A LETTER

"I have missed the last issue of your publication. I have been receiving your magazine since it was first published and have been in the habit of keeping the entire magazine on file instead of clipping portions of it, as I do with most other technical publications. I must have the current issue to keep my files complete and would appreciate your sending me a copy as all my efforts to locate the lost issue at this end have proved fruitless. I might take this opportunity to congratulate you on the high standards you have set so far, and I intend to keep renewing my subscription as long as these standards are maintained."

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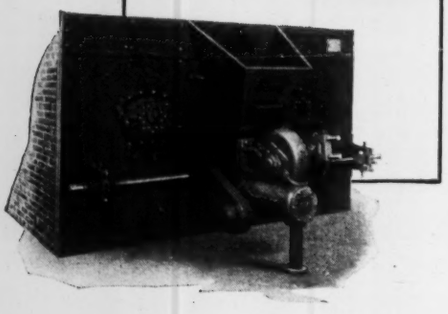

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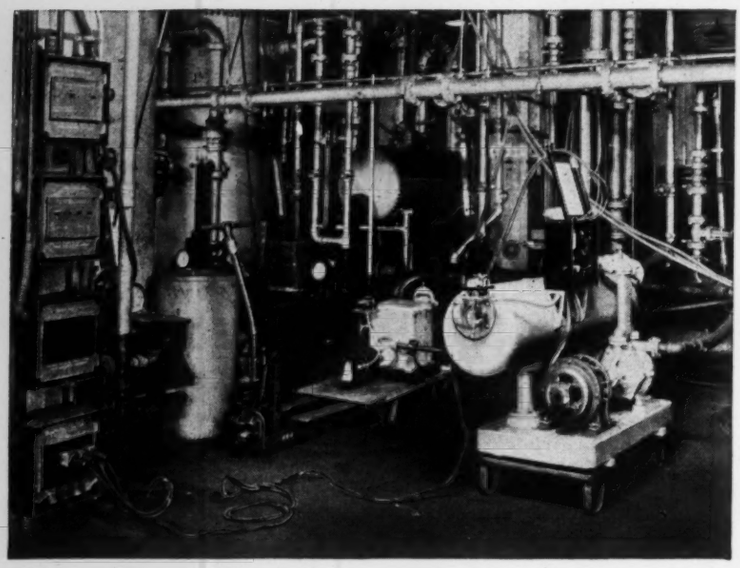

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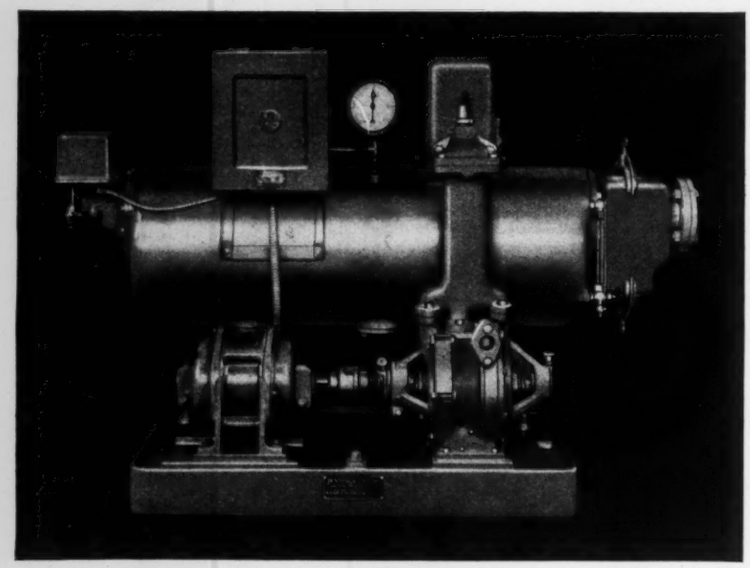

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Heating · Piping and Air Conditioning

Vol. III | July, 1931 | No. 7

Experiences in Air Conditioning Office Buildings

By Samuel R. Lewis*

Because of the inherent characteristics of the two problems, air conditioning for the comfort and health of occupants of an office building presents difficulties not encountered in providing proper atmosphere for manufacturing processes. Samuel R. Lewis has had some interesting experiences in this type of engineering; his views on the subject are presented here.

AIR conditioning for office buildings means, of course, *air conditioning for comfort,* in contra-distinction to air conditioning for manufacturing processes. Air conditioning for manufacturing processes brings about results of such fabulous value that the processes and their exact control have had intensive study and have been well developed by research.

The rooms or spaces in which air conditioning for industry is provided usually can be insulated, and there need not be, generally speaking, any particular fear of drafts, while the spread in temperature between inlets and outlets often may be wide without prejudice.

Problems in Comfort Conditioning

When we approach air conditioning for comfort, however, we cannot use a wide spread in temperature but usually must limit ourselves to only a few degrees difference between the entering air and that in the room.

Office buildings usually are not well protected against sun effect by building insulation and have very large glass areas; doors are in constant use; it is difficult to keep the windows closed. We need not tolerate these conditions in industrial practice, but we must when cooling for comfort. Also, there are both men and women in most offices. The men are always more warmly clothed than the women and so most men are less susceptible to drafts than are most women.

* Consulting engineer, Chicago, Ill.

CEILING DIFFUSERS FOR BOTH INLET AND OUTLET AS USED IN THE AIR CONDITIONED GENERAL OFFICES OF LIBBY, MCNEILL & LIBBY, CHICAGO

CEILING DIFFUSERS FOR INLETS AND OUTLETS AS ROUGHED IN FOR THE A. O. SMITH RESEARCH LABORATORY AT MILWAUKEE

Shafts are Chimneys

Every lofty building has as many different air pressures as a chimney. The stairs and the elevator shafts of such buildings *are* chimneys. There is always a neutral pressure zone in such buildings, below which air leakage is inward, and above which air leakage is outward. Most office building designers, recognizing this, make the elevator shafts as nearly air tight as possible, and isolate the stair shafts from each story with heavy close-fitting doors. If they then fail to provide a properly conditioned supply of air to the lower stories the corridors here will be hot and odorous. If there are any dentists, beauty parlors, cigar stores, and the like on the lower stories, and if any of these have open windows, the rush of air from outside through these windows to serve the air-hungry corridors will carry with it the heart arresting smells of the dentist's room, the objectionable odor of scorched hair from the beauty parlor, and the aroma of the cigar store.

In one thirty-five story tower the chimney effect of the stair shaft, which the law required uninterrupted by doors all the way through from bottom to top, was usually stronger in winter than the powerful exhaust fan provided at the twentieth story for the toilet rooms, to the extent that when the stair shaft doors were opened they back-pulled the toilet vents and caused complaint of odors. In another case the toilet room utility space doors happened to open into a stair shaft. The natural draft of the shaft was stronger than the draft caused by the exhaust fan, and all of these doors had to be weather-stripped and caulked to prevent back drafts.

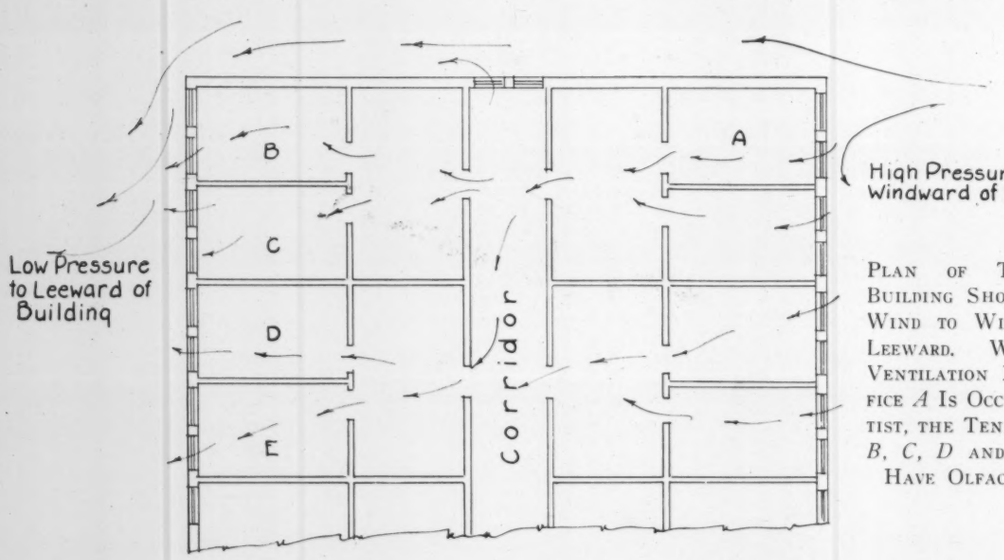
The Window Problem

It is possible to have the windows of an office building fairly tight and they can be fastened shut. The usual tenant, however, hates to be forbidden to open his window. If we have single glass in the windows we must add greatly to the allowances for heat transfer and for infiltration, compared to that necessary if double windows were used.

There are certain difficulties connected with double windows for office buildings. In most cases where double windows are provided the spaces between the two sashes are sealed hermetically. In order to clean the outer glass a special exterior scaffolding runs on a track around the outside of the building. Despite all precautions toward perfect joints between the inner and outer glass there is some infiltration of dust into the dead space, very difficult to remove.

Break System Into Units

It is good practice to have a supply fan with its cold air intake, heaters, coolers, etc., for each story of the modern air conditioned office building, though in some cases each fan may serve five stories or three stories, the fan being located at the center of the vertical zone. When



PLAN OF TYPICAL OFFICE BUILDING SHOWING EFFECT OF WIND TO WINDWARD AND TO LEEWARD. WHEN CORRIDOR-VENTILATION IS USED, IF OFFICE *A* IS OCCUPIED BY A DENTIST, THE TENANTS IN OFFICES *B*, *C*, *D* AND *E* ALL WILL HAVE OLFACTORY EVIDENCE

an attempt is made to serve more than about five stories with each apparatus the ducts tend to become so large as to occupy too much space and there is increased trouble in air distribution due to the weight-temperature differences, when handling cool air.

The Air Inlets

Some system of air distribution for office buildings must be arranged which will be independent of interior partitions, as these inevitably will be changed constantly.

Horizontally-looking overhead side wall inlets may be objectionable as they interfere with furniture, cause drafts, get dirty, and are a problem if it becomes desirable to move the partition.

There are apparently, two alternative inlet points.

One of these is to use the window stool. To reach this with a duct usually requires a suspended ceiling to conceal the duct. It is possible to use the spaces between concrete or skeleton steel joists for these ducts, provided that the structural engineer and the mechanical engineer will cooperate. One very successful ventilating system for an office building uses the spaces between concrete joists, with the corridor ceilings suspended below the transverse supply ducts and concealing also the steam pipes and other services. The corridor ceilings in this building have removable pressed steel panels which look like plaster, to permit access to traps, dampers, valves, etc., of the mechanical equipment.

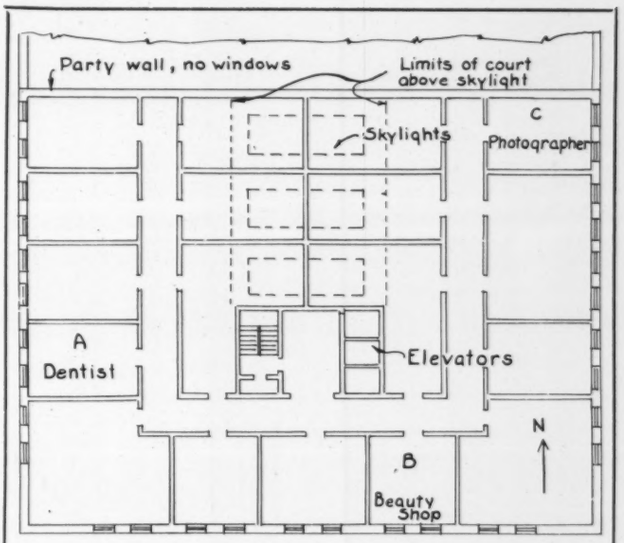
Introduction of air for cooling at the window stools involves careful design. The velocity at the point of introduction must be high, so that the cool air gets up near the ceiling and mixes with the heated air already there before it can touch the people. If the velocity is low the cool air falls over into the room before mixing properly, and causes draft-complaint.

In one recent installation in San Francisco, the air inlets are at one end of each window stool, occupying perhaps one-third of its length while the remaining two-thirds of the space under the window is used for the radiator which heats the room. In this case the supply of air from any single cooling or conditioning system to all of the rooms which it serves is all at the same temperature, but there are several separate systems, one for each general exposure of the building.

With this arrangement the air delivered to the sunny side of the building may be cooler than that delivered at the same time to the shady side of the building. This scheme cannot, however, answer the difficulty often encountered that the executive in his large private office may be shivering while his secretaries in their crowded quarters may be far too warm.

The second alternative air-inlet point is in the ceiling, preferably at the center of each bay; the air being deflected against a flat panel so as to be delivered at comparatively high velocity in a horizontal direction close to the ceiling. Usually with such a device in the center of each bay the flexibility of interior partitions can be coped with as effectively as with window stool inlets.

There is no more difficulty in reaching the ceiling inlets with ducts than there is in reaching the window stool inlets with ducts. Inlets of this type have been used for private office cooling-for-comfort with success even where the story heights were very shallow. The diffusers can be combined with the lighting fixtures and can be made of metal or of ornamental plaster, or of acoustic tile.



PLAN OF AN OFFICE BUILDING IN WHICH THE OWNER FINALLY WAS COMPELLED TO INVEST THREE YEARS INCOME FOR ALL SECOND STORY OFFICES IN INSTALLING A MECHANICAL VENTILATING SYSTEM. STAIRS AND ELEVATORS WERE ENCLOSED. THERE WERE NO OUTSIDE WINDOWS FOR THE CORRIDORS AND THERE WAS NOTHING TO CAUSE ANY AIR CHANGE OR CIRCULATION IN THE CORRIDOR OR IN THE ROOMS UNDER THE SKYLIGHTS. OCCUPATIONAL ODORS FROM THE TENANTS IN OFFICES *A*, *B*, AND *C* PERMEATED THE WHOLE AREA, DEPENDING ON THE WIND-DIRECTION

Use of Corridors as Return Ducts

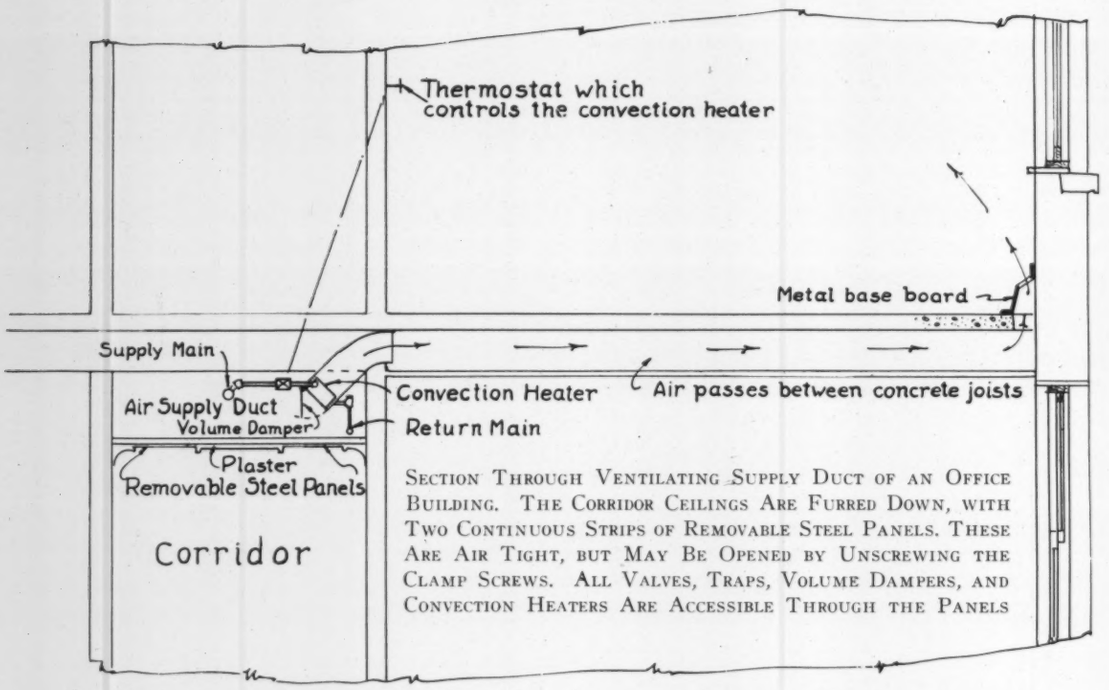
In an endeavor to simplify and to lower construction costs, many existing air conditioning plants for offices make use of the corridors as return air ducts, with the rooms venting through grilles into the corridors, which convey the air back to the supply fan suction for recirculation. This practice makes vent outlets of all leeward rooms, preventing a square deal, air considered, for the occupants of all leeward rooms. This is because, all possible effect of fans to the contrary, there must always be a plus atmospheric pressure on the windward side of every building, and a minus atmospheric pressure on the leeward side of every building. If the windward and the leeward rooms are thrown into open communication across the corridor the outside pressure attempts to equalize itself and the leeward rooms must receive air from the windward rooms.

It may require a fire disaster to cause enactment and enforcement of legislation to prevent general adoption of corridor ventilation, but many engineers warn and protest against the use of corridors and stair shafts as vents.

Requirements of Office Air Conditioning Plants

Having outlined some of the conditions which confront the designer of an air conditioning plant for an office building let us set forth some of the fundamental requirements of such a system.

The air must be introduced at not more than 10 to 12



Section Through Ventilating Supply Duct of an Office Building. The Corridor Ceilings Are Furred Down, with Two Continuous Strips of Removable Steel Panels. These Are Air Tight, but May Be Opened by Unscrewing the Clamp Screws. All Valves, Traps, Volume Dampers, and Convection Heaters Are Accessible Through the Panels

degrees dry bulb cooler than the air already in place.

It must be possible to introduce air at different effective temperatures to different rooms at the same time, both in winter and in summer.

If the windows cannot be double and if the cool walls cannot be insulated, so that they will not get cooler than the dew point of the air, provision must be made for gutters to carry off the water condensed from the air, or the air must be prevented from reaching such a relative humidity as will cause deposition of dew.

The rooms must each have their own independent inlet and outlet, regardless of the corridor. Air may be recirculated, but provisions must be made for direct venting of toilet rooms and for direct venting of rooms which may have tenants who carry on businesses which cause objectionable odors.

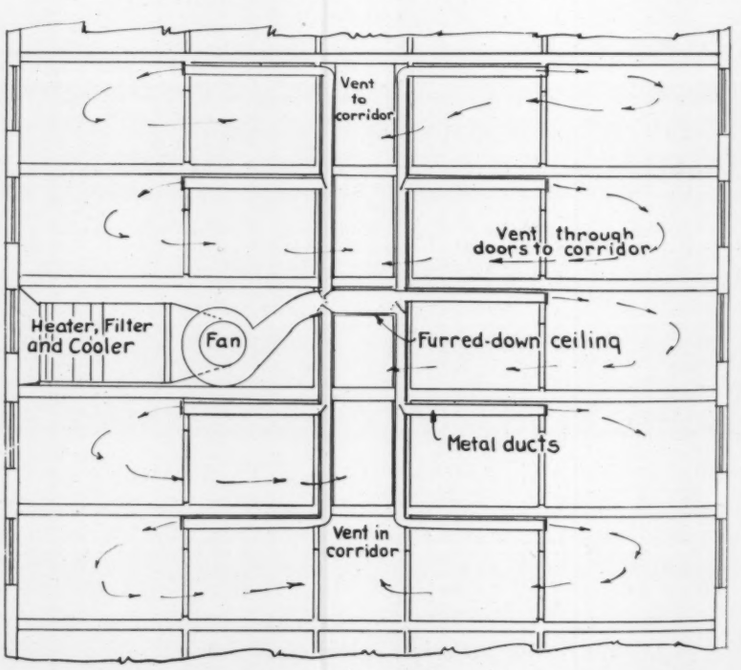
The inlets and the outlets for air must be so arranged that partitions may be shifted within all reasonable limitations without leaving any rooms inaccessible to supply and exhaust openings.

One System Which Meets These Requirements

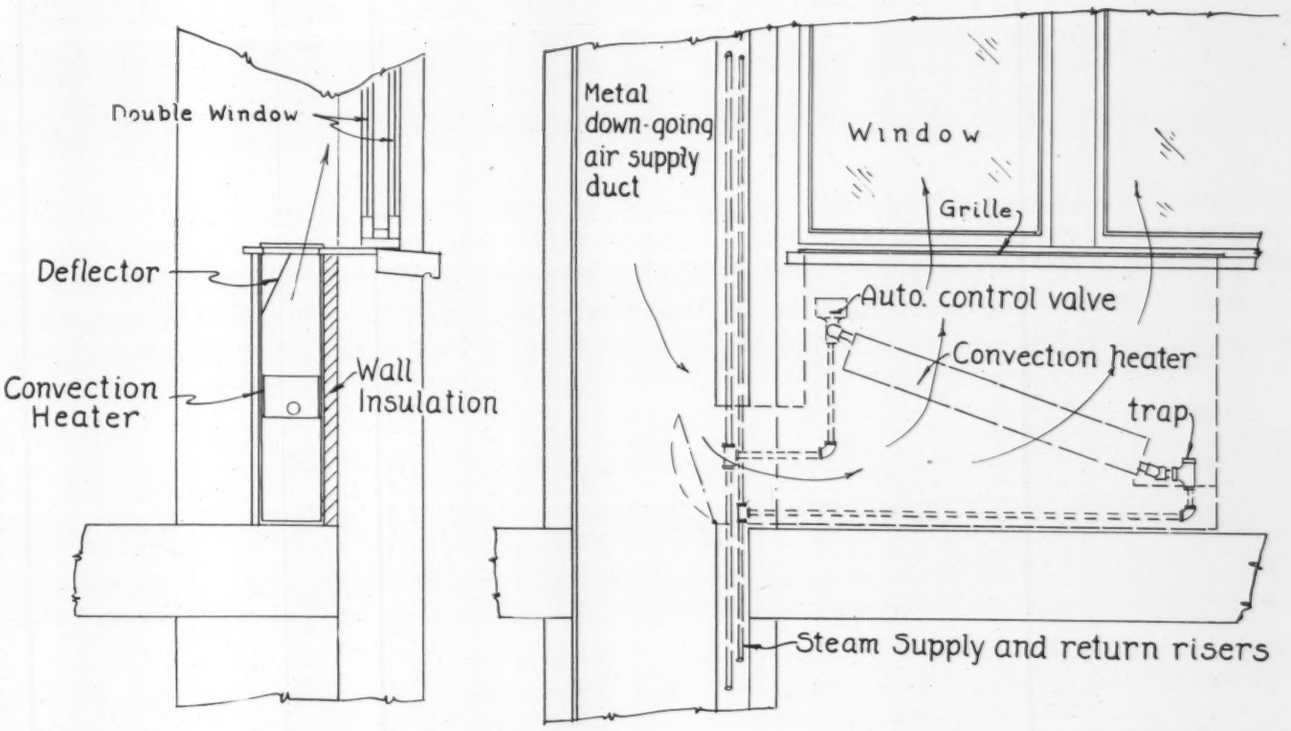
The air within each room in this plant is recirculated by a local electric fan delivering upward through an enclosed convection type heat transfer surface at the window stool. This surface is provided with forced circulation water which is heated in winter and which is cooled in summer. The thermostats in each room control the rate of water flow and so cause the optimum dry bulb temperature of the air delivered by the recirculating fan. By a simple throw-over arrangement the thermostats in the rooms control the water temperature in each radiator in summer as well as in winter.

The water circulated in the summer is never cooler than the dew point of the air which touches the water cooled surface, so that there will be no difficulty about condensation on the heat transfer surfaces.

Each bay, or each room of size smaller than a bay, has a combination ceiling diffuser and exhaust grille, located at about the center of the bay, and having supply and



Section Through a Typical Ventilating System for an Office Building Showing a Supply Fan Which Serves Five Stories. The Furred-Down Corridor Ceiling May Be Awkward When Large Enough for Five Stories and the Distributing Ducts Leading from it Take Up Valuable Floor Space. In One Office Building This General Scheme Was Used, Except that the Ducts Are Above the Structural Floors, in the Fill, Rather Than Below the Structural Floors



Section and Elevation of Window Stool Air Inlets and Convection Heaters in Office Building of Swift & Company, Chicago. When Cooling-for-Comfort, in This Case, if the General Office Air Supply Is Too Cool for a Private Office, Steam Is Turned on the Convection Heater Shown; by This Means a Measure of Control in Summer as well as in Winter Is Obtainable. The Deflector Below the Grille Was Found Necessary to Prevent Draft Complaint When a Low Velocity Delivery of Cool Air Was in Effect

exhaust ducts between the joists from laterals run below the joists in the furred-down corridors.

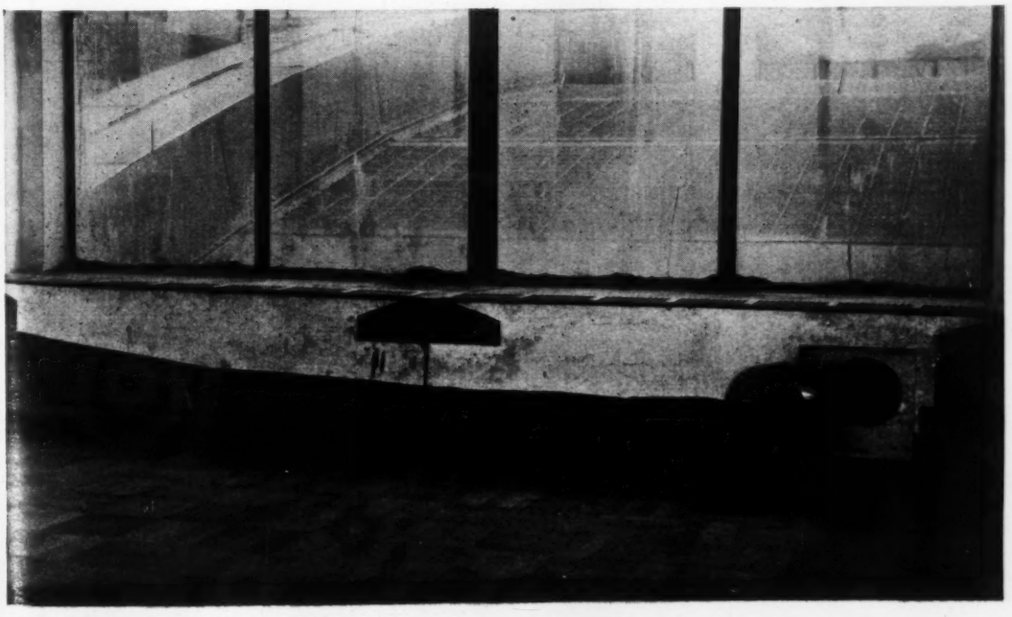
The combination winter and summer heat transfer surfaces in the rooms are so large, due to the low temperature spread of the cooling function, that the winter hot water is comparatively low in temperature. It is not essential to operate the recirculating fans in winter.

The air supply to the ceiling diffusers and the exhaust from this point is only large enough to serve the needs of ventilation in keeping down odors, and the air withdrawn from the ceiling outlets may be used for flushing out the corridors and toilet rooms on its way to the toilet exhaust fans, or may be exhausted directly, or may be recirculated.

The air introduced at the ceiling inlets is dehumidified in summer and humidified in winter, is dust-free, and need not necessarily be very much different as to dry bulb temperature from the air already in the room.

It is desirable that this air shall be slightly cooler, both winter and summer, than the air already in the room.

So potent is the cooling-by-evaporation service of the human body that a wide range of effective temperature comes within the comfort zone. We may dehumidify the air by cooling it with a chilled water spray below the dew point and condensing out the water, or may absorb the water from the air into a moisture-hungry sponge, like silica gel. Both types of dehumidifying appa-



Local Convector at Window Stool Which Uses Warm Water in Winter and Cold Water in Summer, with Electric Recirculating Fan

ratus are available commercially and both types are in successful use.

A plant along these lines has been built for the A. O. Smith Corporation. It should give better results as to control and as to operating cost than any plant which is compelled to use air all at the same cool temperature or all at the same warm temperature for many rooms on different exposures or orientations, and this plant with its duct-enclosed circuits is expected to be more satisfactory than plants which use the corridors as vent flues.

Cooling One Private Office

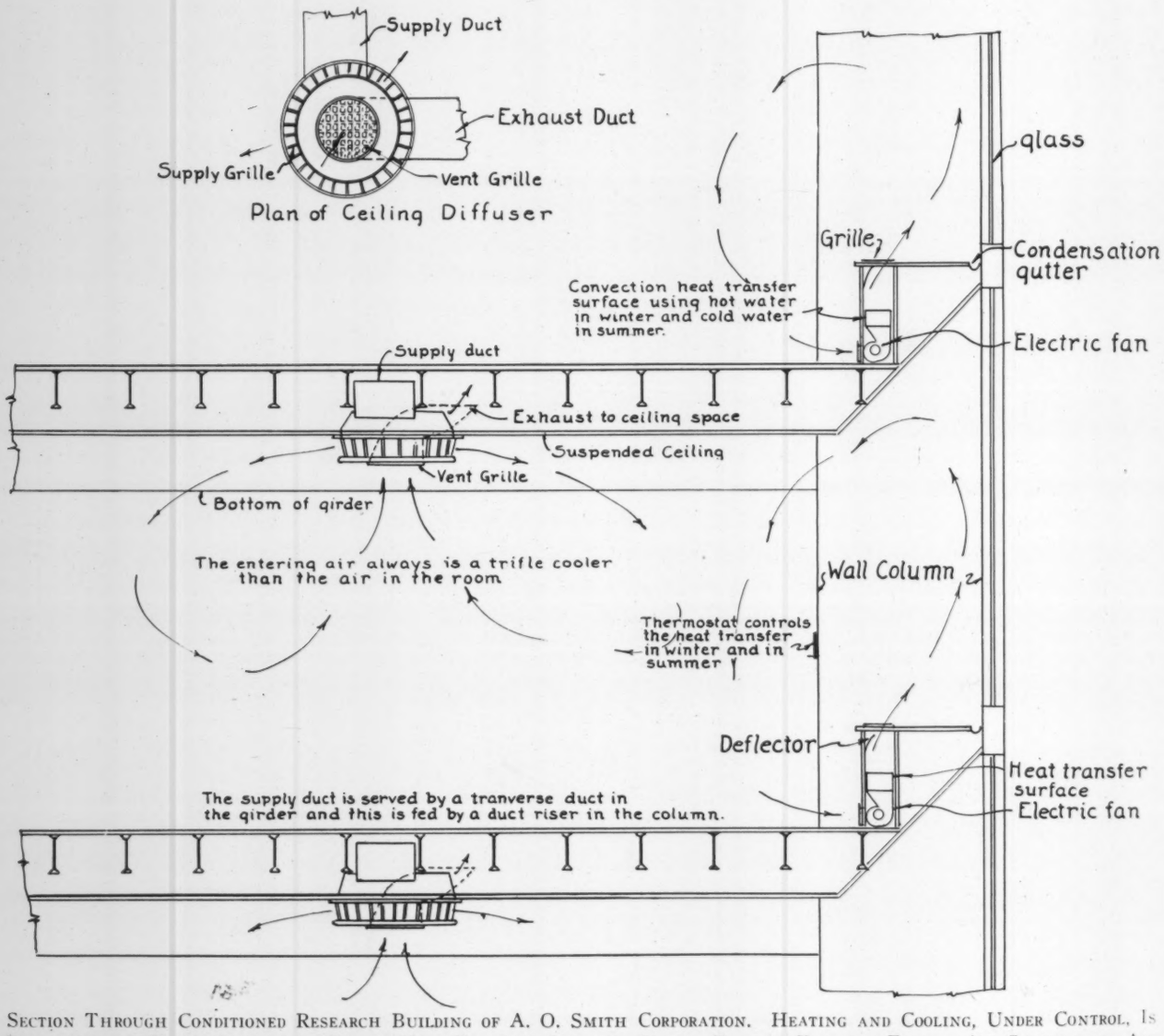
Local fan-equipped unit coolers supplied with chilled water or which use direct expansion refrigerants are new, but are available and practicable. One may install a compressor somewhat larger than that required for the ordinary large ice box cooler, in the basement, serving from it a fan-equipped unit cooler which may stand on a pedestal in the private office, promoting wonderfully pleasant conditions in a very simple and easily controlled manner.

These local cooling units are available in sizes as small as one-ton capacity (around 12,000 Btu per hr. in air drying and cooling capacity).

About half of this energy might be required to condense moisture. Of the other half, 1,000 Btu per hr. might be needed to absorb the heat given off by two people, 500 Btu per hr. to care for the lights, and the remainder, 4,500 Btu, to balance the heat gain by sunshine on the exposed walls and glass and by transmission from adjoining uncooled rooms, and by air leakage. It will be understood of course, that this unit plant will serve only one comparatively small private office.

The cost of air cooling for comfort in an office building is given in a recent paper by W. H. Carrier as $1.25 per sq. ft. of floor space, with an over-all annual cost of 19 cents per sq. ft. of floor space and a cost of one cent per man-hour, based on the annual over-all costs of investment and operation.

The Chicago meat packers use air cooling for comfort in most of their general office buildings and have done so for many years. They have found by experience that the cost is well justified in increased efficiency and health of employes.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Exhaust Duct
Supply Grille
Vent Grille
Plan of Ceiling Diffuser
glass
Grille
Condensation gutter
Convection heat transfer surface using hot water in winter and cold water in summer.
Electric fan
Supply duct
Exhaust to ceiling space
Suspended Ceiling
Vent Grille
Bottom of girder
The entering air always is a trifle cooler than the air in the room
Wall Column
Thermostat controls the heat transfer in winter and in summer
Deflector
Heat transfer surface
Electric fan
The supply duct is served by a tranverse duct in the girder and this is fed by a duct riser in the column.

Section Through Conditioned Research Building of A. O. Smith Corporation. Heating and Cooling, Under Control, Is Done by Local Forced-Circulation-Water Heat-Transfer Surfaces Having Electric Fans for Forcing Air Circulation. Air Purity and Moisture Content Are Controlled by Introduction and Removal of Conditioned Air at the Ceiling at the Center of Each Bay

High Pressure Steam and the Modern Industrial Plant

By E. H. Barry*

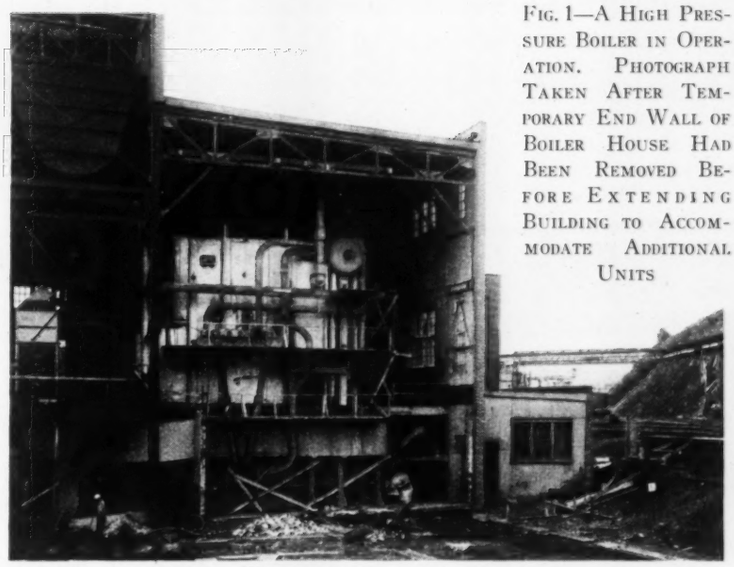


FIG. 1—A HIGH PRESSURE BOILER IN OPERATION. PHOTOGRAPH TAKEN AFTER TEMPORARY END WALL OF BOILER HOUSE HAD BEEN REMOVED BEFORE EXTENDING BUILDING TO ACCOMMODATE ADDITIONAL UNITS

The question of steam pressure and temperature for the industrial plant is one of importance to the PIPING ENGINEER; this article is a review of the considerations leading to the choice of steam pressure where process steam is needed and power is generated as a by-product. While the article does not go into the factors directly concerned with the determination of pipe sizes, layouts, etc., it discusses the work preliminary to that, and points out the method by a typical example

FEW readers can have failed to notice the marked trend in recent years toward high pressure steam power plants. The movement was accelerated in 1925 when the Edgar Station of the Edison Electric Illuminating Co. of Boston went into operation. This station stands as a pioneer among high pressure power plants in this country, but since then the movement toward higher pressures has gained steadily in momentum. Today there are over half a dozen central stations in operation at pressures of 1200 lb. and higher. More than thirty *central stations* operate at pressures between 300 lb. and 1200 lb., we understand.

The year 1926 witnessed the construction of several high pressure *industrial plants*. There are now nearly one hundred such plants operating at pressures of 400 lb. and upwards.

*Supervisor of Power, West Virginia Pulp & Paper Co., New York City.

"In order to reduce the total cost of power, the ratio of by-product power to total power must be increased. This ratio may be increased by:

Increasing the demand for low pressure steam.
Lowering the pressure of the process steam.
Improving the efficiency of non-condensing prime movers.
Raising the initial steam pressure."

Prior to 1925, the average large central station obtained a kilowatt-hour from about 20,000 Btu at the coal pile. The central station today, through the use of high initial steam pressures and temperatures, and the regenerative and reheat cycles can produce a kilowatt-hour with less than a pound of coal, the conversion rate closely approaching 13,000 Btu per kilowatt hour.

Opportunities for Economies When Process Steam Is Used

Industries which use large quantities of steam in process present opportunities for effecting phenomenal economies in the generation of power. Before the advent of high pressure boilers and bleeder turbines, it was common for even large manufacturing concerns to generate power by means of straight condensing turbines, and to supply process steam direct from boilers. The heat equivalent of power so made often ran as high as 50,000 Btu at the coal pile for each kilowatt-hour. Today, with bleeder turbines available, designed to meet the particular conditions of a given plant, and with the use of modern steam generating equipment, it is possible for many industrial plants to generate at least a part of the electrical power at the low rate of 5000 Btu per kilowatt-hour. The explanation lies in the difference between the operating conditions of the central stations and the particular types of industrial plants in which these economies may be effected.

A large group of industries use heat in their processes, and when the necessary temperatures are not too high, this heat can be utilized in the form of steam. In this category are pulp and paper mills, chemical plants, soap factories, laundries, textile mills, and a host of others. While in many industries the cost of steam and power constitutes but a small fraction of the total manufacturing cost of the product, the annual coal bill in many plants runs as high as—or higher than—a million dollars. The possibility of savings on the order of 50 per cent in the coal bill warrants serious consideration on the part of plant executives and engineers. Economies of this order have actually been accomplished in many industrial plants during the past five years by the installation of modern high pressure power plants.

By-product Power

In designing a modern power plant for an industrial organization, the aim of the power engineer is to approach the ideal condition in which all the required power is generated by the steam used in process. If this goal is reached, the cost of the power so made will be remarkably low, each kilowatt-hour being produced at the expense of about 5000 Btu released at the coal pile. Since power is produced by the expansion of steam from boiler pressure down to the pressure used in process, it follows that the greater the demand for process steam and the greater the difference between boiler pressure and process pressure, the greater the amount of power produced by the process steam. Power so produced is usually known as *by-product* power. As will be seen later, the coal equivalent of a unit of by-product power does not depend on the thermodynamic efficiency of the prime mover. That is to say, the coal cost of by-product power is the same, whether made in an inefficient reciprocating engine or in the most highly refined and efficient turbine. Neither does the cost depend on the steam pressure. These apparently anomalous statements require some explanation.

Secondary Power

The *amount* of by-product power produced by a given quantity of process steam does, of course, vary with the efficiency of the prime mover and with the initial and final steam pressures. If the total power demand is greater than the amount of by-product power made, the balance of the power requirements must be made up from another source, frequently at a cost greater than that of the by-product power. Such power we may call *secondary* power.

Reducing Cost of Total Power

Secondary power may be purchased from a source outside the plant; it may be generated by condensed steam; by Diesel or other internal combustion engines; or by hydraulic engines. The cost of secondary power, except in the case of hydro, is usually greater than the cost of by-product power. In order to reduce the cost of total power, the ratio of by-product power to total power must be increased. This ratio may be increased by:

Increasing the demand for low pressure steam.
Lowering the pressure of the process steam.
Improving the efficiency of non-condensing prime movers.
Raising the initial steam pressure.

There would obviously be no economy in the extravagant use of process steam merely for the sake of increasing the amount of by-product power. It is often possible, however, to increase the consumption of process steam by changing certain process methods, working up waste products, increasing production, or by supplying neighboring plants with low pressure steam for heating or other purposes.

Increased Pipe Sizes May Be the Answer

Much may be done in the way of reducing the pressure of the process steam in the average plant. In numerous instances, increasing the size of steam piping has effected

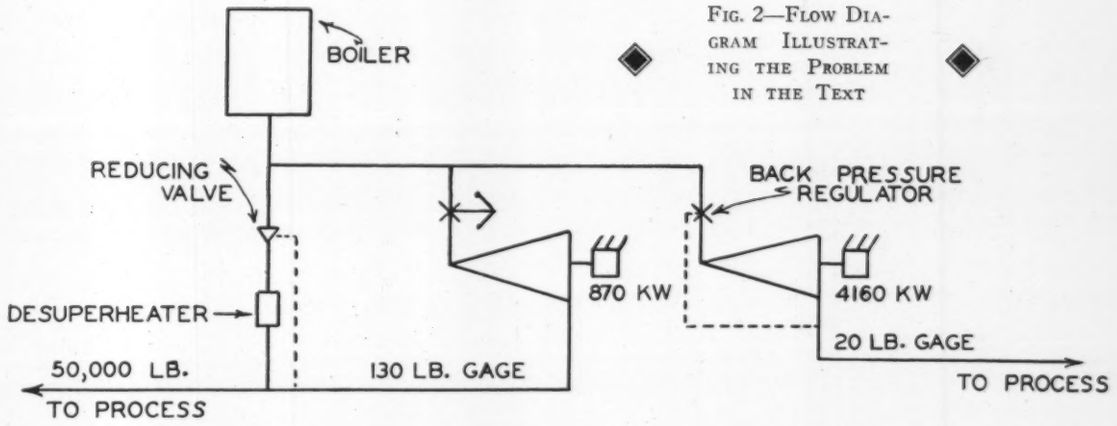


FIG. 2—FLOW DIAGRAM ILLUSTRATING THE PROBLEM IN THE TEXT

economies which have paid for the piping changes the first year. By reducing the pressure drop in the steam piping, a lower exhaust pressure can be maintained in the power plant, thus increasing the amount of by-product power generated. Again, there are many processes using steam at relatively high pressure which could equally well use steam at far lower pressure. In some cases the determining factor is the temperature and not the pressure. In others, it is the total heat quantity, irrespective of either temperature or pressure, which is important. If means can be found to utilize steam at lower pressures, the gain in reduced power costs will usually pay for the necessary process changes.

Replace Inefficient Prime Movers

The replacement of inefficient prime movers with others of higher efficiency will increase the amount of by-product power. In some cases the replacement of steam drives with motor may bring about a reduction in the cost of power, although each case of this sort requires special study. It frequently happens that the electrical transmission and conversion losses over-balance any gain in efficiency of prime movers.

Or, Perhaps High Pressure Steam

Finally, after the three foregoing possibilities have been investigated, there remains the question of generating steam at high pressures. This is undoubtedly the age of high pressure steam. Boiler manufacturers are prepared to make steam generators for practically any pressure. Research will doubtless evolve alloys which will successfully withstand considerably higher temperatures than those of today, in combination with high pressures.

Necessity for Re-heating

Equipment is available for temperatures to 750 F in combination with pressures of 1800 lb. per sq. in. With higher temperatures, the use of the re-heat cycle could be avoided. The need for re-heating when high initial steam pressures are used is an obstacle in the way of adoption of extremely high pressure by the average industrial plant because of the cost of the re-heating equipment; the carrying charges on the investment may offset the savings due to increased thermal economy.

If the initial steam pressure is sufficiently high so that all power generated is by-product power, there will, of course, be no need for a condensing turbine. Expansion of the steam will stop when the lowest process pressure is reached. In a case of this kind, the necessity for re-heating is determined partly by the amount of moisture tolerable in the turbine and partly by the requirements of the process in regard to the quality of the process steam. Some drying processes, for example, are accelerated by the use of dry saturated steam in place of superheated steam. This is due to the high rate of heat transfer between the steam and a wetted metal surface. Other processes are of such a nature as to require dry steam.

Process Steam a Conveyor

When steam is used in process, it serves as a conveyor of heat. Usually the substance of the steam is not consumed in the manufacturing process. The manufacturing departments require delivery of heat by the power plant at a given rate. The number of pounds of steam to be delivered, therefore, depends on the heat content of each pound. This fact must be considered when the design of a high pressure power plant is contemplated. If there are special requirements as to the pressure or temperature of the process steam, these facts, too, must not be overlooked.

Supplying Process Steam for a Typical Plant

In order to illustrate the application of some of the principles described above, consider the problem of determining the necessary pressure at which steam must be generated in a typical industrial plant, fixing the primary condition that the process steam shall generate all the necessary power, without the necessity of rejecting any heat to power house condensers. The steam demands are:

50,000 lb. per hr. at 130-lb. gage dry saturated (no objection to superheat).
100,000 lb. per hr. at 20-lb. gage, 5 F superheat.
The power demand is 5,000 kw.

All of these demands are variable, particularly the demand for power and 130-lb. steam. We will assume that the amounts given represent the average demands throughout the year and arrive at an approximate solution, later correcting the result to meet the actual conditions.

Since the quality of the 20-lb. steam is specified, this demand will constitute the starting point of the solution. The heat content of 20-lb. steam at 5 F superheat is 1169 Btu per lb. The requirement of 50,000 lb. per hr. of 130-lb. dry saturated steam is more conveniently expressed in terms of heat units. Each pound of 130-lb.

dry saturated steam contains 1193 Btu. Thus, the 130-lb. demand is for 50,000 × 1193 or 59,650,000 Btu per hr.

Use of Mollier Diagram

For the sake of simplicity, assume that two simple non-condensing turbines will be used for the generation of power, both taking high pressure steam at their throttles; one exhausting at 130 lb., the other at 20 lb. Assume also that the Rankine cycle efficiency of each unit is 70 per cent, considering the overall performance of the turbine generator units in terms of power delivered at the switchboard. By means of the Mollier diagram, this problem can be solved graphically. The initial condition of steam necessary for the fulfillment of the above conditions is 340-lb. gage, 170 F superheat.

It is possible to solve a problem of this type analytically. The author has worked out a general expression for the initial state point of steam necessary for a balance between process steam and power demands with given conditions similar to those described in the present problem. The application of the method, however, is a tedious process involving, finally, the solution of a second degree equation. The graphical method is quicker and the solution can be found with only a few trials.

Referring to Fig. 3, showing a portion of the Mollier heat entropy chart, the first step is to select trial points on the pressure line marked 35 *lb. abs.* Each trial point represents the theoretical condition of steam exhausted at 20-lb. gage from a turbine with 100 per cent Rankine cycle efficiency; in other words, one which converts 100 per cent of the available heat into power.

Suppose the first trial point is taken at the intersection of the 35-lb. pressure line and the 1110 Btu total heat line. The actual heat of the 20-lb. steam desired is 1169 Btu per lb. The difference between the actual desired heat of the steam and the heat content of the exhaust of the ideal turbine (1169 — 1110 = 59) known as the reheat, is 30 per cent of the available heat (since the assumed efficiency of the machine is 70 per cent). Therefore, the available heat from throttle state point to exhaust is 59 ÷ 30 per cent, or 196 Btu.

Adding 196 to the total heat at the trial point (1110), we get 1306 Btu as the total heat of the throttle steam corresponding to the trial point. By proceeding vertically upward on the Mollier chart from the trial

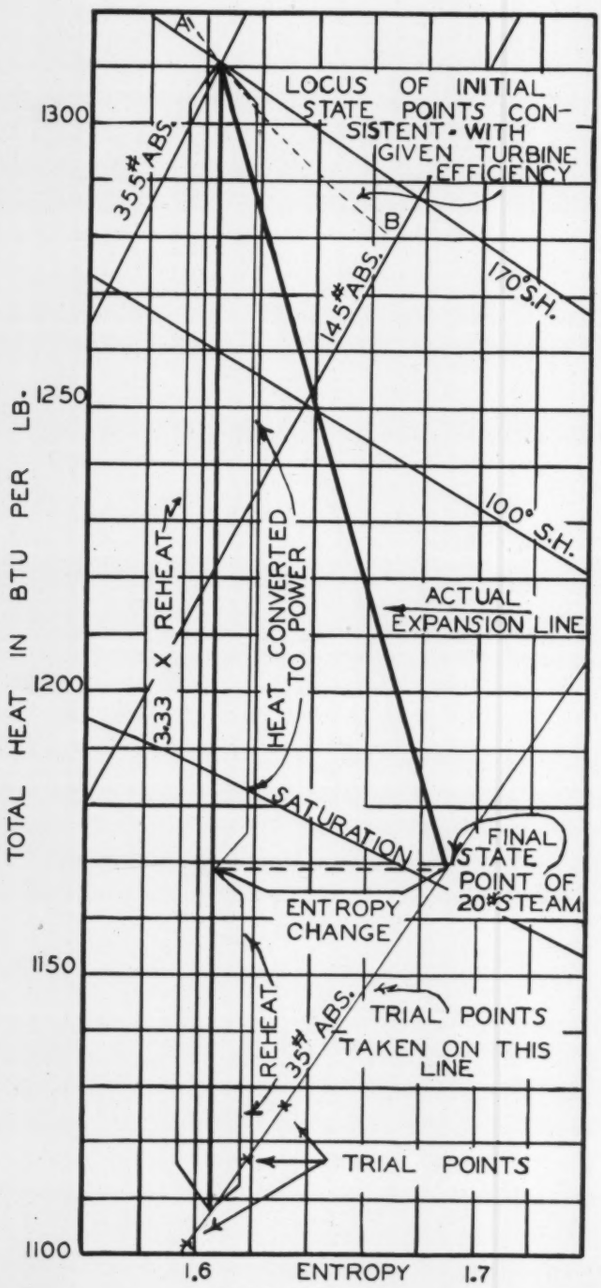


FIG. 3—MOLLIER DIAGRAM SHOWING THE GRAPHICAL METHOD OF FINDING INITIAL STATE POINT OF STEAM TO SATISFY CONDITIONS OF PROBLEM IN TEXT

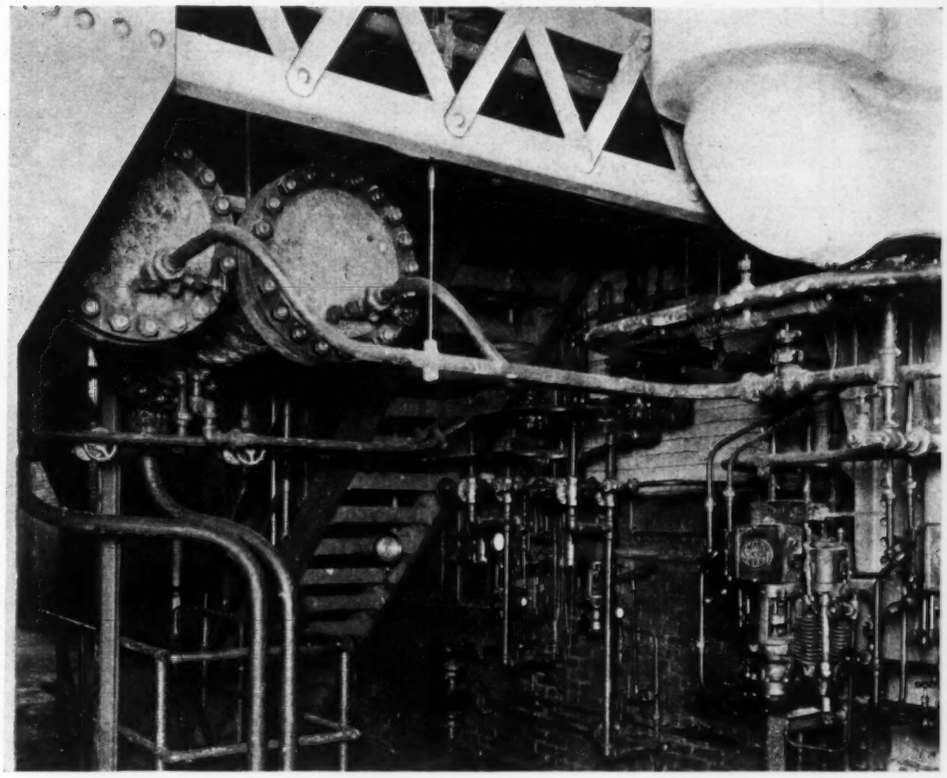


FIG. 4—AUTOMATIC CONTROL BOARDS FOR REDUCING AND DESUPERHEATING HIGH PRESSURE STEAM

point to the horizontal line representing a heat content of 1306 Btu we read the pressure and total temperature of the throttle steam.

We can now check back and determine whether throttle steam at this pressure and temperature, in the quantities demanded by the process, will develop the necessary amount of power. First, take 70 per cent of the available heat to 20 lb. ($.7 \times 196$) which gives 137 Btu per lb. of 20-lb. exhaust which is converted to power. By dividing 3415 (the heat equivalent of 1 kw.-hr.) by 137, get the water rate of the turbine exhausting at 20 lb., which is 24.9 lb. per kw.-hr. Since this turbine exhausts 100,000 lb. of steam per hr., it develops $\frac{100{,}000}{24.9} = 4020$ kw.

Now determine the power developed by the turbine exhausting at 130 lb. Starting from the state point of the throttle steam and descending the same vertical line as before, stop at the 130-lb. pressure line (145 lb. abs.) and read the heat content as 1224 Btu. The available heat from throttle to 130-lb. exhaust is therefore 1306 — 1224 = 82 Btu. 70 per cent of this amount of heat (57 Btu) is converted to power, hence the actual heat content of the 130-lb. exhaust steam is 1306 — 57 = 1249 Btu per lb. The water rate of this turbine will be

$$\frac{3415}{57} = 59.9 \text{ lb. per kw.-hr.}$$

Since the process requires 59,650,000 Btu per hr. at 130 lb. and since each pound of 130-lb. steam contains 1249 Btu, the process will require 47,800 lb. per hr. of 130-lb. steam. The turbine will develop $\frac{47800}{59.9} = 800$ kw. The two turbines will have a combined output of 4020 plus 800 or 4820 kw., which is less than the necessary amount of 5000 kw. This result indicates that other trial points must be taken.

Table 1 shows the results of three trials, the third trial giving 5030 kw. as the combined output of the two machines, which is near enough to 5000 kw. for practical purposes. By spotting the initial state point at the intersection of the horizontal line representing a total heat of 1311 Btu with the vertical line through the final trial point, we can read the pressure and superheat direct from the chart. These are found to be 340 lb. gage (355 lb. abs.), 170 F superheat.

TABLE 1

	Trial 1	Trial 2	Trial 3
Actual heat at 20 lb	1,169	1,169	1,169
Total heat at trial point	1,100	1,110	1,108
Reheat at 20 lb	69	59	61
Available heat 20-lb. turbine $\left(\frac{100}{30} \times \text{reheat}\right)$	230	196	203
Initial heat content	1,330	1,306	1,311
Initial pressure (lb. abs.)	475	320	355
Heat at 130 lb. (ideal expansion)	1,211	1,224	1,222
Available heat, 130-lb. turbine	119	82	89
Heat converted to power, 130-lb. turbine	83	57	62
Water rate, 130-lb. turbine, lb. per kw.-hr.	41.2	59.9	55.1
Actual heat of 130-lb. steam	1,247	1,249	1,249
Lb. 130-lb. steam used per hour	47,800	47,800	47,800
Heat converted to power, 20-lb. turbine	161	137	142
Water rate, 20-lb. turbine, lb. per kw.-hr.	21.2	24.9	24.04
Power from 130-lb. turbine	1,160	800	870
Power from 20-lb. turbine	4,720	4,020	4,160
Total power	5,880	4,820	5,030

"It behooves the wide-awake industrialist to prepare for the replacement of present plant long before its apparent usefulness has been outlived. By so doing, he is prepared to cope all the more successfully with the increasing difficulties of modern competition."

Process Demands Not Constant

Having found the pressure and temperature at which steam must be generated in order that the two turbines shall generate 5000 kw. with a flow of 50,000 lb. per hr. through the machine exhausting at 130-lb. back pressure, and a flow of 100,000 lb. per hr. through the unit exhausting at 20 lb., it next becomes necessary to consider the fact that the process demands for steam and power are not constant, but vary between certain limits. Since it is desirable to eliminate the need for condensers, the steam pressure must be sufficiently high to satisfy the maximum demand for power which can exist at times of a minimum demand for process steam.

Allow for Future Trends

Naturally an analysis of the momentary demands for power and steam requires a vast amount of time and, after its completion, there still remains the uncertainty of the future trend of these demands. If the plant expands, increasing its production, it is possible that the demands for power and steam will increase in the same proportion, in which case it is necessary only to increase the capacity of the steam and power generating units. If, however, as is usually the case, the demands for power increase at a greater rate than the demands for process steam, it is necessary for the designer to prepare for this condition by selecting a steam pressure high enough to insure the production of the necessary power at all times in spite of a relatively smaller demand for steam.

May Require Higher Pressure

It is necessary, therefore, to go above the balancing pressure, and since the required investment in power plant equipment will not increase in direct proportion to the pressure, it may prove feasible to go to 400 lb. or even 500 lb. with a corresponding increase in the total temperature of the steam. In this example, it is quite likely that the selection of 500-lb. pressure boilers would be justified.

Reducing Valve and Desuperheater

Provision must be made for passing steam through a reducing valve and desuperheater in order that the process may be supplied with the necessary steam at times of reduced demand for power, when, of course, the turbines will exhaust insufficient steam to satisfy the process demands.

The problems of regulating the flow of steam through the turbines and reducing valve to maintain a constant frequency of the electric current, and constant pressures in the process steam lines, require painstaking study on the part of the designer. Much development work remains to be done by manufacturers of control apparatus, but continual progress is being made. Plant engineers can assist in this work by placing their problems in the hands of the manufacturers of this type of equipment.

There is no royal road to the attainment of the ideal industrial power plant. Certain operating difficulties increase when high steam pressures and temperatures are employed for the generation of power. Standardized high pressure equipment cannot yet be picked off the shelves of the manufacturer's warehouse. The progress made during the last few years in the development of modern power plant equipment presages more remarkable progress for the future. It behooves the wide-awake industrialist to prepare for the replacement of present plant long before its apparent usefulness has been outlived. By so doing, he is prepared to cope all the more successfully with the increasing difficulties of modern competition.

Two Valuable Articles Appear Next Month

Piping engineers will find Harvey A. Wagner's discussion of the theory of design for piping a valuable part of the next issue of HEATING, PIPING AND AIR CONDITIONING. In it, the author considers the Clavarino, Birnie and Barlow formulas; important phases of the problem are described clearly and are fully illustrated with diagrams.

How to save money in motor operation is the subject of Louis Mackler's article which will appear in the August issue of HEATING, PIPING AND AIR CONDITIONING. In it, the author shows just what over- or under-voltages mean in regard to the cost of driving fans, pumps, etc., by motors; he also describes the remedies.

Air Conditioning of Press Rooms†

A Review of Requirements, Advantages, Costs

By I. C. Baker*

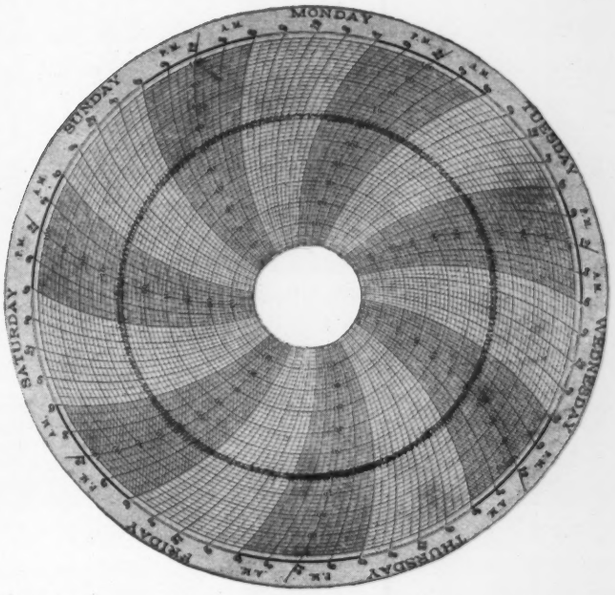


FIG. 1—TYPICAL CHART FROM HUMIDITY RECORDER SHOWING MAXIMUM VARIATION OF FIVE PER CENT RELATIVE HUMIDITY DURING ONE ENTIRE WEEK'S OPERATION

MAINTAINING constant and desirable air conditions in the press room is a problem of considerable importance to every printing and lithographing plant in that variations in temperature and humidity result in troubles which materially reduce production, increase costs and interfere with high quality work. Air conditioning may, therefore, be considered as a production problem and its value determined on the basis of lower printing costs.

The deformities of paper such as wavy edges, buckling, curling, creasing, stretching and shrinkage caused by variations of atmospheric conditions in the press room are production problems which can be minimized or entirely eliminated by maintaining uniform and correct temperatures and humidities.

Static, offset, roller troubles, variations in ink, dust and other press room problems causing non-productive hours for the presses are also usually caused by *improper* air conditions, and can be definitely controlled by *proper* air conditions.

Object of This Discussion

The object of this paper is to discuss briefly—

The physical properties of air.
The variations which exist in the outside atmosphere and in unconditioned press rooms.
The effect of variations in atmospheric conditions on paper.
The effect of variations in air conditions in the operation of the press room.
The effect of temperature and humidities on the health and comfort of the press men.
Proper and economical temperatures and humidities in the press room.
Data on initial and operating costs of air conditioning or humidifying systems.

Interesting data on the initial and operating costs of air conditioning systems with and without refrigeration are included in I. C. Baker's paper on air conditioning press rooms.

In this article, the author shows what air conditioning can do in improving production and decreasing costs in printing plants and discusses the methods of doing it. Industrial air conditioning is growing so that this is an article to be studied with care.

† From an address at 44th Annual Convention of United Typothetae of America. (Appeared in *Typothetae Bulletin*, November 10, 1930.)

* York Ice Machinery Corporation, York, Pa.

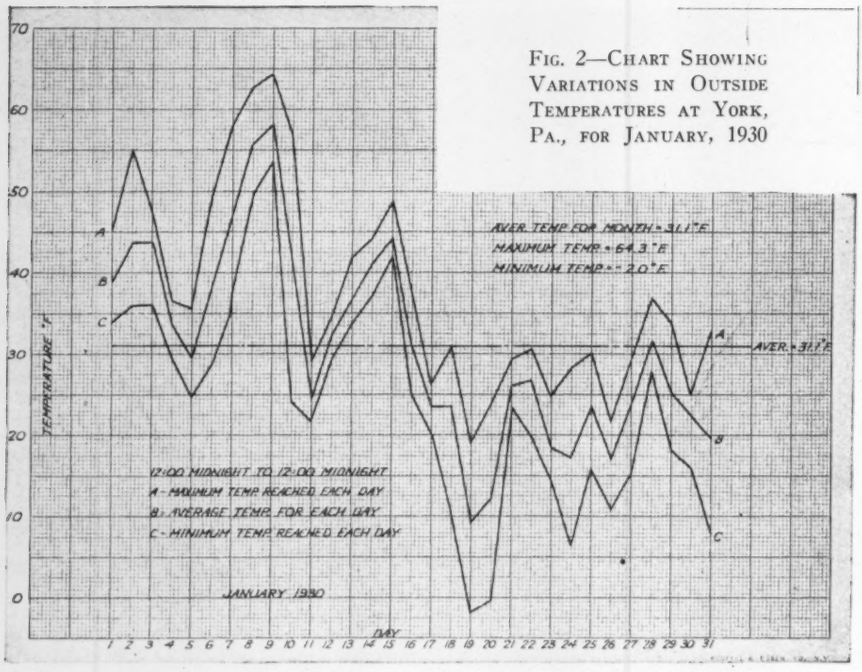

AVER. TEMP FOR MONTH = 31.1°F.
MAXIMUM TEMP. = 64.3 °F.
MINIMUM TEMP. = -2.0 °F.
AVER. = 31.1°F.
12:00 MIDNIGHT TO 12:00 MIDNIGHT
A - MAXIMUM TEMP. REACHED EACH DAY
B - AVERAGE TEMP. FOR EACH DAY
C - MINIMUM TEMP. REACHED EACH DAY
JANUARY 1930
TEMPERATURE °F.
DAY

FIG. 2—CHART SHOWING VARIATIONS IN OUTSIDE TEMPERATURES AT YORK, PA., FOR JANUARY, 1930

. . . . . if relative humidities must be maintained constant at some given point, humidity control is quite essential in summer as well as in winter and presents a rather complex problem to the air conditioning engineer if he is to maintain constant conditions at all times of the year.

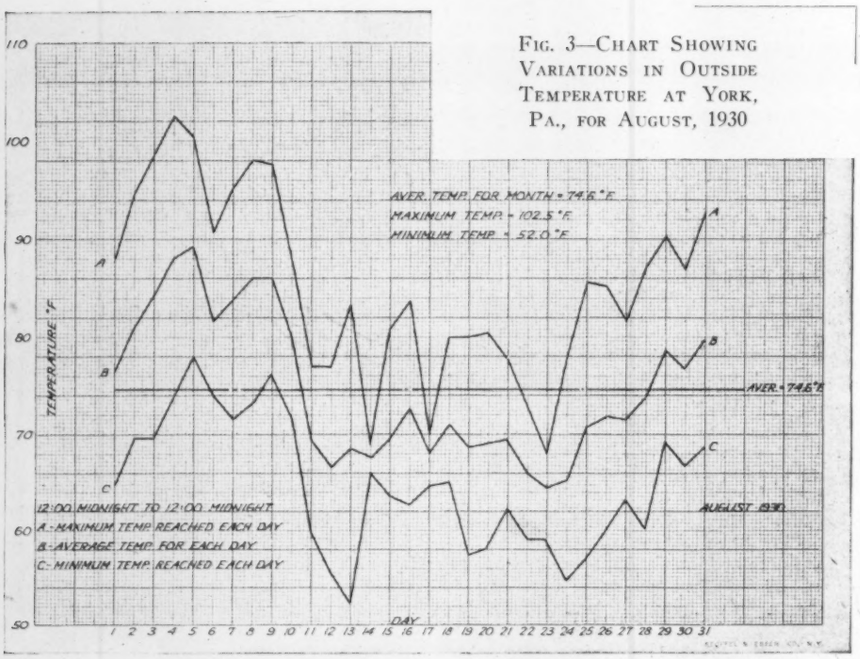

AVER. TEMP. FOR MONTH = 74.6 °F.
MAXIMUM TEMP. = 102.5 °F.
MINIMUM TEMP. = 52.0 °F.
AVER. = 74.6°F.
12:00 MIDNIGHT TO 12:00 MIDNIGHT
A - MAXIMUM TEMP. REACHED EACH DAY
B - AVERAGE TEMP. FOR EACH DAY
C - MINIMUM TEMP. REACHED EACH DAY
AUGUST 1930
TEMPERATURE °F.
DAY

FIG. 3—CHART SHOWING VARIATIONS IN OUTSIDE TEMPERATURE AT YORK, PA., FOR AUGUST, 1930

The Physical Properties of Air

Air as we usually find it is essentially a mixture of oxygen and nitrogen containing a variable amount of moisture or water vapor and also impurities in the form of vapors, gases and finely divided solid particles, usually referred to as dust.

We are all familiar with the fact that air may be either hot or cold and we usually judge the temperature of the air by our personal state of comfort or by consulting the ordinary thermometer. The *temperature* of the air in the press room *in the winter* may be kept at a satisfactory point by means of various types of heating systems. The *moisture content* of the air, however, usually referred to as its humidity, is of prime importance in the press room and unfortunately varies over a wide range.

Air can hold only a certain amount of water vapor at a given temperature. For instance, at zero degrees Fahrenheit, which is a common outdoor temperature in the winter, air is capable of containing approximately ½ of a grain of moisture per cu. ft. (a grain of moisture being 1/7000th part of a pound) whereas at 70 F the same air is capable of containing approximately 8 grains per cu. ft. Relative humidity may be defined as the amount of moisture actually in the air compared to the amount it is capable of containing when saturated or when it holds all the moisture it is capable of holding at any particular temperature. Air when saturated is said to have a relative humidity of 100 per cent.

Necessity of Humidification

From the facts just stated one may readily see why we have such low humidities in the press room without humidification in cold weather. If outside air at zero degrees and 100 per cent relative humidity either leaks in or is blown into a press room, containing ½ grain per cu. ft., which is all the moisture it is capable of containing, and when in the press room has a temperature of 70 degrees and is then capable of containing 8 grains per cu. ft. it is obvious that the air in the press room will be only ½ divided by 8 or 1/16th saturated or will have a relative humidity of 6¼ per cent; an extremely dry condition.

Thus it can be seen that air is not actually dried when heated as water vapor has not been removed but rather the capacity of the air to hold moisture has been increased and thus the air at the higher temperature and the correspondingly lower relative humidity will absorb moisture more readily from paper or any hygroscopic material, such as rollers, with which it comes in contact.

Variations in Outside Air

The author has kept a continuous record for the past year of dry and wet bulb temperatures by means of a recording dry and wet bulb thermometer which reveals interesting facts. During January, 1930, the range in outdoor temperatures was from a maximum of 64.3 F January 9 to a minimum of —2 F on January 19. The greatest range in this month for a 24-hour period or single day is from a maximum of 58 F on January 10 to a minimum of 21 F on January 11. This change of temperature is capable of producing changes in relative humidity in the unconditioned press room from 65 per cent to 6 per cent during the month and variations of more than 40 per cent in relative humidity during a single day.

During the month of August, 1930, the total range of outdoor temperatures varied from 102.5 F on August 4 to 52 F on August 13 with variations of relative humidities ranging from 17 per cent to 90 per cent.

From this data it is possible to see, if relative humidities must be maintained constant at some given point, that humidity control is quite essential in summer as well as in winter and presents a rather complex problem for the air conditioning engineer if he is to maintain constant conditions at all times of the year.

In the press room, the heat given off by the presses and occupants together with the heat added by the sun effect on walls, windows and the roof, further adds to the temperature in the press room in summer and thus to the moisture carrying capacity of the air.

The Effect of Variations in Atmospheric Conditions on Paper

Paper is essentially cellulose made principally from wood, cotton and linen to which is added filler to make the paper opaque and smooth and a sizing of glue, rosin or starch to add to its strength and make a better surface to hold the ink.

Paper absorbs or gives up moisture very readily depending principally on the relative humidity of the air with which it comes in contact. As the paper absorbs moisture the individual fibers swell, thus causing the entire sheet to become larger and wider. Due to the fact that the fibers align themselves in the direction of movement of the paper machine as the paper is manufactured and also due to the fact that they swell more in diameter than they increase in length the greatest stretch is at right angles to the paper machine travel.

Paper will absorb moisture in proportion to the relative humidity of the air surrounding the sheet and, therefore, its dimensions will vary with the relative humidity of the air. The average change of dimension of a 38-in. sheet across the machine with a change of 30 per cent in relative humidity will be approximately one-sixth of an inch and usually about one-twelfth of an inch in the direction of travel of the machine. This change of dimension which, as has been shown may occur during a single day, is sufficient to destroy the register where multi-color work is being run.

The center of a pile of paper is under heavy pressure and it is practically impossible for the air to penetrate into the center of the pile. In case the paper in the pile absorbs moisture the outside edges of the paper increase in length compared to the center of the paper, thus causing the wavy edges which make feeding of the presses difficult. In case the paper dries at the edges due to a decrease in the relative humidity, it will be found that the outside edges of the paper will shrink, thus causing the paper to bulge in the center. The bulging of paper also makes feeding difficult and is likely to cause creasing as the paper goes through the press. It will, therefore, be seen that in order to prevent wavy edges and buckling of the paper that it must be held at somewhat near the same conditions at which it is received from the mill.

Paper Manufacturers Must Air Condition Plants

Although paper manufacturers give considerable attention to the weight of the paper as it leaves the machine they have given little attention to the proper air

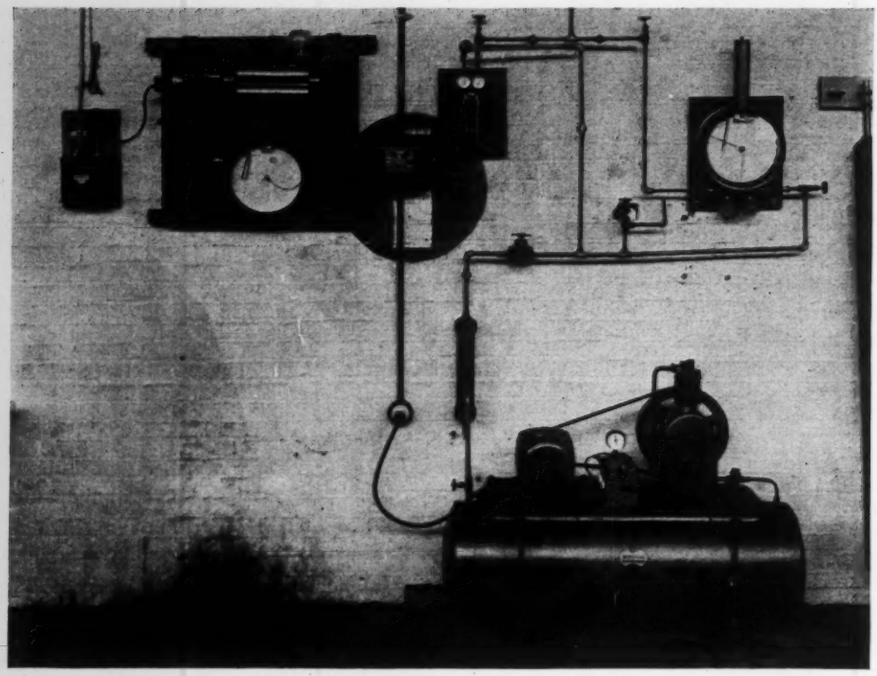


FIG. 4—HUMIDITY CONTROL APPARATUS AND RECORDING DRY AND WET BULB THERMOMETER

The author suggests that a committee composed of printers, lithographers, paper manufacturers and air conditioning engineers be given the problem of determining the relative humidity which should be carried uniformly in all press rooms, in order that paper manufacturers might send out their paper normal to one definite humidity.

FIG. 5—UNIT TYPE AIR CONDITIONER IN PRESS ROOM WITH AUTOMATIC CONTROL

conditions surrounding the paper after it leaves the machine and has passed into the calender room, finishing rooms, and stock rooms. If these rooms are not maintained at the proper humidities the paper will lose or gain moisture before it is packed, and therefore, may reach the printer with varying moisture content.

It will probably never be possible to entirely eliminate the effect of wavy edges, shrinking and creasing, unless the paper manufacturers air condition their finishing, calender and store rooms, so that the printer can depend on uniform moisture content of the paper as received.

There is an increased interest on the part of paper manufacturers in air conditioning systems for their plants. Recently several installations have been made in a large paper mill in Pennsylvania with automatic control of the temperature and humidity for the finishing and calender departments.

If the paper manufacturer would properly condition his plant so as to have a uniform moisture content of the paper and if the stock and press rooms in a printing plant were conditioned for the same relative humidity, it can readily be seen that the annoyances from paper distortions before the paper is put on the press would be eliminated.

Cooperation Needed

If constant conditions are maintained continuously many of the production problems in the press room are eliminated. However, it is not going to be possible to eliminate entirely all of the troubles due to paper distortion until there is a standard set for press room operation whereby all press rooms will be maintained at approximately the same relative humidity and until the paper manufacturers co-operate to the extent of making all of their paper normal to the same conditions which have been selected for the press room. I would like to suggest at this time that a committee composed of the printers, lithographers, paper manufacturers and air conditioning engineers be given the problem of determining the relative humidity which should be carried uniformly in all press rooms, in order that the paper manufacturers might send out their paper normal to one definite humidity which would be universally adapted to air conditioned press rooms, regardless of the kind of paper.

Effect of Variations in Air Conditions on the Operation of Press Rooms

Static electricity is a press room problem resulting in sheets adhering to each other, offsetting and making necessary expensive slip sheeting and loss of time on the presses. Static electricity is generated by the friction of paper as it goes through the press. Dry air being almost a perfect insulator prevents the paper from discharging its electricity, with the result that the printed sheets are attracted together at the delivery end, pressing the wet ink against the next sheet and thus causing offset. Moist air being a conductor of electricity permits the sheet to lose its charge and thus permits an air cushion to form between the sheets that eliminates offset, permits more rapid drying of the ink and allows the paper to be more easily handled. Neutralizers eliminate some of the trouble due to static and are helpful even in the conditioned press room, but do not solve the problem as completely as does maintaining proper humidities.

Rollers, being a mixture of glue and glycerine, absorb or give up moisture due to variations in relative humidity the same as paper. Absorbing moisture in high humidity air, they increase in diameter as much as one-quarter of an inch, causing the roller to ride the form, fill up the plates and result in dirty jobs with the necessity of frequent washups and loss of time. Frequent adjustment of rollers is also necessary where variations of relative humidity occur. The tack of the rollers is also affected by the moisture content and when the rollers become too wet they lose their affinity for the pigment of the ink. Proper and constant air conditions in the press room lengthen the life of rollers and permit the use of one grade of rollers throughout the year.

Destruction of register due to change of dimensions of the sheet between runs is a direct result of changes of relative humidity and causes costly delays for re-registering or waiting until the humidity condition changes.

High humidities and the resulting high moisture content of the paper have a decided effect on the absorption of the ink. Wet paper retards absorption and causes spreading; paper too dry causes too rapid absorption and too deep penetration of the ink.

Changes in *temperature* have little effect on paper but do affect press room operation, particularly the condition of the ink and rollers. Most inks flow best at temperatures from 70 to 80 F. This range of temperature is also best for the rollers.

Effect of Temperature and Humidities on the Health and Comfort of the Press Men

The effect of temperatures and humidities in the press room on the health and comfort of the employes is another factor in production, as there can be no denying the fact that comfortable employes turn out more and better work. By producing comfortable and healthful conditions labor turnover can be reduced to a minimum and lay-offs due to sickness avoided.

An extended study of atmospheric conditions to promote human comfort has been conducted by the *American Society of Heating and Ventilating Engineers* in conjunction with the *U. S. Bureau of Mines* at its laboratory in Pittsburgh. Effective temperatures have been determined experimentally which express a person's feeling of warmth or cold for various combinations of temperatures, humidity and air motion. Generally, higher humidities should be accompanied by lower temperatures for the same degree of comfort.

Proper and Economical Temperatures and Humidities in the Press Room

In winter it has been found that most persons slightly active are most comfortable at an effective temperature of 64 F, which corresponds to a temperature of approximately 67½ F at 50 per cent relative humidity, although the range of the comfort zone extends from 65 to 74 F at 50 per cent relative humidity. For press room conditions the upper part of the range should be used with temperatures preferably held between 70 and 72 F. These temperatures, with a corresponding relative humidity of 50 per cent, are best adapted as far as production is concerned and are satisfactory from a health and comfort standpoint.

During the summer the temperatures producing maximum comfort vary with the outside temperatures. When

FIG. 6—CENTRAL STATION SYSTEM MAINTAINING PROPER ATMOSPHERIC CONDITIONS FOR FINISHING AND PRINTING FINE STATIONERY

a temperature of 95 F exists outside an inside temperature of 80 F will be found comfortable. Likewise at 85 F outside, a 77 F temperature will be comfortable, while at 75 F outside a temperature of approximately 73 F will be comfortable.

Where press rooms are to be held at a constant temperature during the summer season a temperature of 80 F is satisfactory from the comfort standpoint and also economical from an operating standpoint as far as the air conditioning system is concerned.

Initial and Operating Costs of Air Conditioning or Humidifying Systems With and Without Refrigeration

It is a complex problem to determine initial and operating costs, due partly to the variations in the nature of the buildings, which determine to a great extent the size and capacity of the air conditioning equipment which must be installed. Press rooms with large exposed roof areas and skylights have a larger heat load due to the sun effect in the summer time and greater exposure in the winter time. The spacing of presses in the press room is also a variable which makes it impossible to apply rules for determining operating costs and fixed investment generally.

Another factor which varies the cost of air conditioning equipment is the load factor. It will usually be found that the actual current consumption is about one-third of what might be expected when considering the total horse-power installed on the presses.

However, several jobs have been analyzed with both unit air conditioners and with central systems and it has been found that a volume of air will have to be supplied which will give an average of a ten-minute air change for distribution purposes and also to absorb the heat which is generated and flows into the room from the outside. On this basis the operating and fixed charges of the air conditioning system with and without refrigeration reduced to cost per 1,000 cu. ft. of space conditioned can be approximated.

There has been found very little difference in the initial cost between unit type of air conditioning systems and the central type of air conditioning systems, although this statement is not true for all types and kinds of buildings as the construction costs vary over quite a range, depending on where the equipment is to be located and the structural changes necessary in the building itself. Air conditioning systems with air washers or humidifiers of the unit or central type, including automatic control and including the necessary radiation for heating the building will usually vary from $20.00 to $30.00 per thousand cubic feet of space with an average of $25.00 for press rooms having a content of approximately 150,000 cu. ft. For very small press rooms the cost may reach $100.00 per thousand cu. ft. From this figure should be deducted (in a new press room) the cost of installing the heating system which the air conditioning system replaces.

For the sake of comparison, and to point out the method, we will assume that the average allowance for direct steam radiation would be approximately 25 sq. ft. of radiation per 1,000 cu. ft., and we will assume that the probable cost is in the neighborhood of $0.75 per sq. ft., thus making the cost of the heating system approximately $18.75 average per 1,000 cu. ft. of space heated, exclusive of the boiler and main piping which would be common in both systems. When this is deducted from the average cost of the air conditioning system, it will be found that the air conditioning system would cost $6.25 a thousand cu. ft. of space in addition to the usual cost of the heating system.

In one plant analyzed it has been found that a press room containing 12 cylinder presses has a content of 180,000 cu. ft. Dividing the cubical contents by the number of presses gives 15,000 cu. ft. per press. The initial cost of an air conditioning system without refrigeration would then be approximately $375.00 per press, or if the cost of the heating system is deducted (according to our assumption) a cost for air conditioning of approximately $94.00 per press. The total cost of the air conditioning system, *including refrigeration,* in the particular case cited above, would be approximately $750.00 per press.

On the same basis the air conditioning system without refrigeration would have an operating cost due to power consumption, on the basis of an electrical current rate of 2c per kw.-hr., of approximately one-tenth of a cent per 1,000 cu. ft. space per hour.

If refrigeration were added to this system the additional cost of the refrigeration during the season of the year that it was operated would be approximately three-tenths of a cent per 1,000 cu. ft. of space per hr. This estimate is based on using a cooling tower for the condenser. It is often customary to estimate depreciation,

obsolescence and interest on the equipment of this type on the basis of 16 per cent per year which allows 6 per cent for interest and 10 per cent for depreciation and obsolescence.

On the basis of the entire investment without deducting for the heating system, or approximately $375.00 per press as the initial cost of the air conditioning system without refrigeration, it can be seen that the fixed charges per press per year would be $60.00. Dividing this yearly fixed charge by the number of hours per year gives an hourly fixed charge per press of 0.685c. Again on the basis of 15,000 cu. ft. of space per press and at the rate of one-tenth of a cent per hr. per 1,000 cu. ft. of space, it will be seen that the operating expense will be one and five-tenth cents per hour per press. The total fixed and operating costs would then be approximately 2.185c per hour per press.

Considering an air conditioning system equipped with refrigeration with a total initial investment of $750.00 per press, the yearly fixed charge on the basis of 16 per cent would amount to $120.00. However, in figuring the hourly rate, one should take into consideration the number of days the refrigerating machinery would run which would vary depending upon the climate in the particular location of the plant. Assuming 150 days' operation at 10 hours a day, or 1,500 hours operation per year, the fixed charge would amount to 8c per hour per press for the actual operating season.

The operating costs during the time the refrigerating plant was in operation would be .4c×15, or 6c per press hour.

Figuring the fixed charges on the basis of the actual operating time as just stated where we arrived at a fixed charge of 8c per press hour and 6c per press hour for operating current, it can be seen that the entire cost of the system for summer operation would be 14c per press hour whereas, in the winter time when only the air washer was in operation, the cost would be reduced to 2.185c per press hour.

The fixed and operating charges have been reduced to press hours in order that there might be a comparison with the cost of running the press which would probably run from $4.00 to $6.00 per hour.

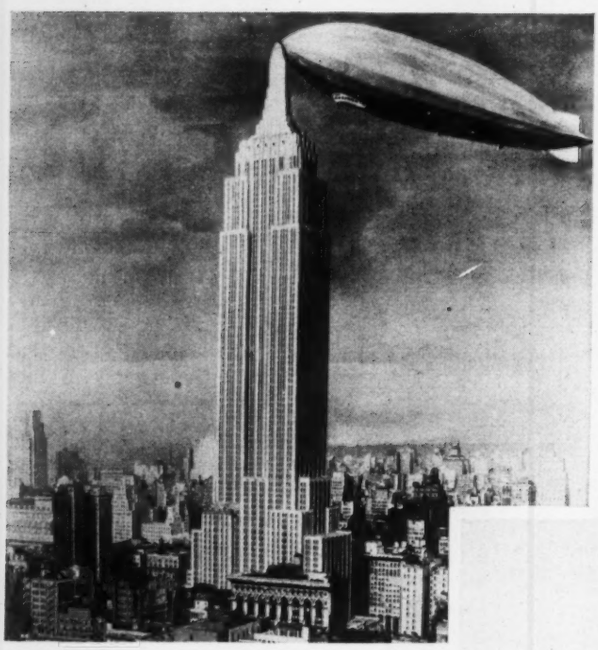
Savings

From this comparison it is seen that the cost of the air conditioning system in comparison to operating the press is very small, in fact almost negligible, and if the air conditioning system would make the presses more productive or say, if an appreciable amount of time that the presses might be shut down due to the lack of an air conditioning system were saved, that it would take comparatively few hours of non-productive time on the presses to pay for the operation of the air conditioning system.

Another way of looking at this problem is that if an air conditioning system would make a $6.00 an hour press 10 per cent more productive the air conditioning system at 14c an hour would show a profit of 46c per hour per press during the summer and approximately 58c per press hour during the winter.

There are other things to be taken into consideration that are difficult to measure in dollars and cents: goodwill of the custmers due to better work, increased volume of business due to high class work, reduced labor turnover, the health and comfort of the employes and many other factors of a similar nature which should be credited to an air conditioning system which would greatly increase the profit that one might expect from its operation.

Baltimore Conditions



THE NEW 86-STORY EMPIRE STATE BUILDING IN NEW YORK CITY REACHING A QUARTER OF A MILE ABOVE 5TH AVE. AT 34TH ST., SHOWING HOW A DIRIGIBLE WOULD LOOK MOORED TO THE 200-FT. MAST

Air conditioning engineers should be particularly interested in the announcement of the Baltimore and Ohio Railroad that its train, "The Columbian," between New York City and Washington, D. C., is air conditioned from head end to observation end. Starting in service May 24 this is a progressive step in the control of temperature and humidity for human comfort and health. The forerunner of the "comfortable Columbian" was the "Martha Washington," the dining car conditioned in the spring of 1930. Undoubtedly, other roads will follow suit—several are now operating conditioned diners and wide interest is shown by railroad men.

THE CLUB LOUNGE CAR WHICH IS INCLUDED IN THE EQUIPMENT OF THE COMPLETELY AIR CONDITIONED "COLUMBIAN"

& Ohio Air
Its "Columbian"

COLONIAL - TYPE DINING CAR IN SERVICE ON THE B. & O.'S COMPLETELY AIR CONDITIONED "COLUMBIAN"

Each car has its own air conditioning unit . . . windows remain closed, for cleanliness and quiet . . . . automatic control of temperature and humidity . . . . the first train in history to be air conditioned completely.

THE BALTIMORE AND OHIO "COLUMBIAN," FIRST COMPLETELY AIR CONDITIONED TRAIN IN THE WORLD, LEAVING WASHINGTON, D. C. THE "COLUMBIAN" IS OPERATED DAILY IN BOTH DIRECTIONS BETWEEN NEW YORK CITY AND WASHINGTON

A NEW AND STRIKING VIEW OF THE CAPITOL AT WASHINGTON

By means of improvements in the white water piping system, the Port Huron Sulphite & Paper Company has reduced its loss of good stock from about 8 to 5 per cent. Also, the quality of the stock has been improved, sources of dirt being eliminated through careful piping design. This article tells the story of this excellent example of what proper industrial piping can do to production and costs

Paper Mill Improves Product *through* Piping Changes

By Max W. Benjamin

IN LINE with an extensive program of mill improvements the Port Huron Sulphite and Paper Company began last year the revamping of their screen room white water system.

The system is now complete, and is a valuable improvement, both from the standpoint of efficient use of the white water and in the cleanliness of the paper stock produced. The loss of good stock has been reduced from about eight to five per cent, and the quality of the stock has been visibly improved, due to the fact that sources of dirt have been eliminated by careful design.

Use of "White Water"

In a sulphite pulp mill a large quantity of water is used as the medium for conveying the fibers and fiber bundles from point to point in the process of refining, which is necessary to the production of high quality paper stock. Eventually most of the water is separated from the stock by passing the mixture of stock and water over a screen of fine mesh which catches the fibers, allowing the water to pass. Normally such screens have seventy or more meshes per inch. The water so removed always contains some good fibers, and frequently carries large amounts of valuable chemicals which have been added at various points in the process. This water is called "white water" and great economy results from returning to the system as much of it as possible. Loss of good stock is reduced by this procedure, and the cost of water may or may not be reduced, depending on the source of fresh water. If fresh water is pumped from an elevation much lower than the mill, or purchased, a saving in pumping cost will usually result from re-use of white water.

The advantages of re-use of the sulphite mill white water have long been realized, and at Port Huron, a system for returning large quantities of it to the flat screens, where the final refining is accomplished, has been in operation for some years.

New Arrangement Installed

At the beginning of 1930 several portions of the screen room white water system were in need of replacement, and rather than recondition the old system it was decided to install a complete new arrangement. Fig. 1 is a flow diagram of the sulphite mill and shows the white water circuit as it exists in the new system.

The first consideration in the design of the new lay-out was cleanliness. A piping material which would remain clean inside and out was essential.

Piping for this purpose is surrounded by an acid atmosphere most favorable to corrosion. In addition a deposit of a black and pasty nature may form on the inside of some kinds of pipe.

Lay-out of System

The system is laid out as shown in Fig. 2, with a main header at one side of the screen room, hung away from over the screens. An 8-in. header is required, and is made of spiral pipe, leadized after fabrication. This header supplies three 4-in. copper branches which in turn each supply four 2-in. down spouts. There are sixteen flat screens served by this system; four parallel banks with four screens in series in each bank. The 2-in. down spouts add white water to the fiber and water mixture between adjacent screens in each bank.

At one end the 8-in. header receives white water from the removal pumps of either or both of two sulphite lap machines. At the other end the header receives water from the white water pump of the secondary washers. Floating on the header is a tank of 4,700 gal. capacity into which two wet machines in the bleach plant discharge white water. This tank also has a connection for fresh water make-up, which is controlled by a float valve, and a direct connection to the sulphite lap machines mentioned above. An emergency overflow line to the sewer prevents the tank from running over and flooding the screen room. Fig. 2 shows these connections.

From the bottom of the tank an 8-in. leadized line is taken across the screens to the head box of a large paper machine and also to the white water pump of that paper machine. A branch of 6-in. copper tubing runs from this 8-in. line to the primary washers and the knotter head box.

Piping Arranged for Easy Cleaning

In the case of all lines feeding white water into the screens, provision has been made for easy and thorough cleaning of the interior of the pipes. Each end of each line terminates in the run of a tee, or equivalent, which is blanked off. All that is necessary for inspection or cleaning of a line is removal of the blind flanges at each

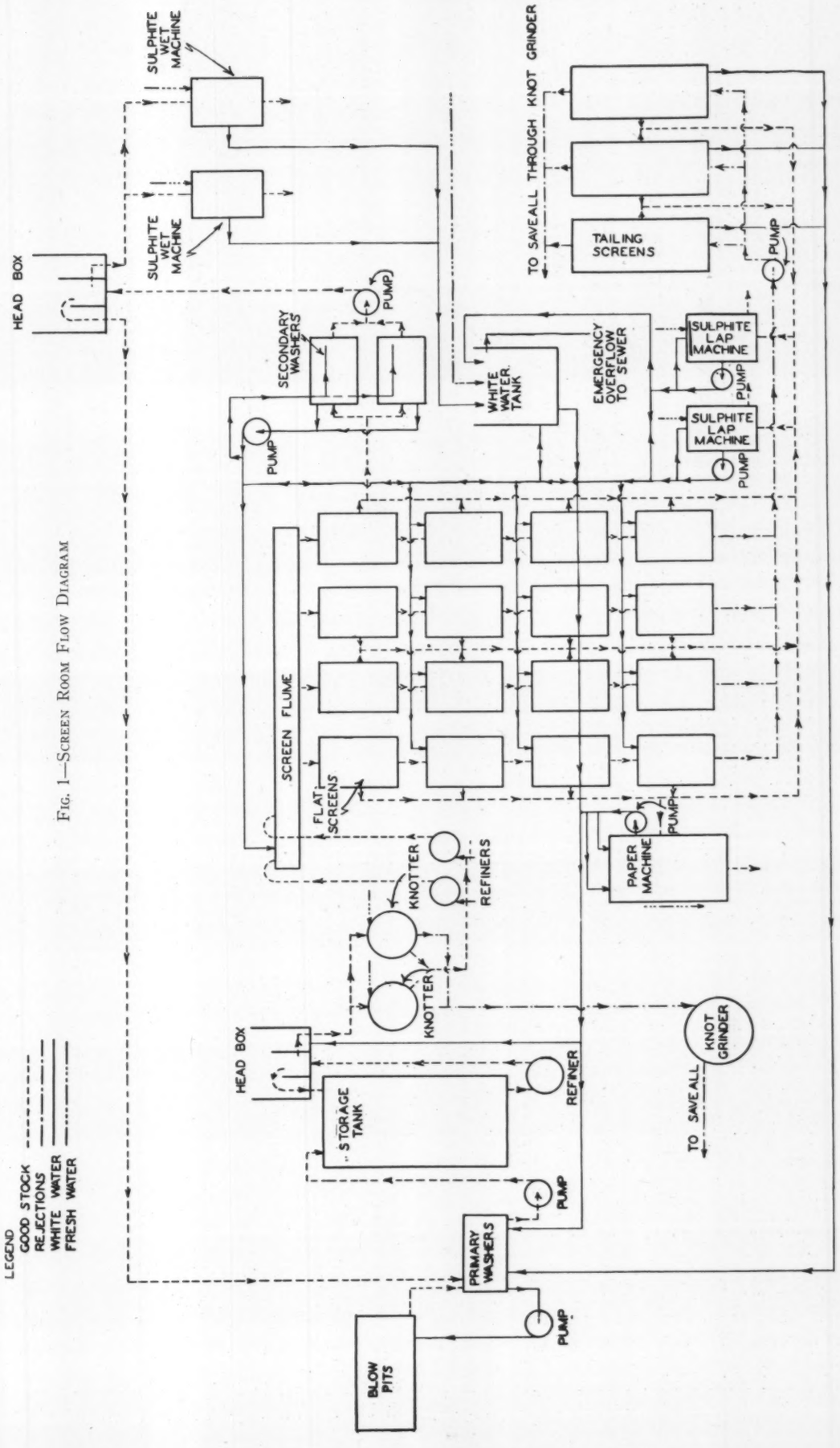


FIG. 1—SCREEN ROOM FLOW DIAGRAM

end of the line; it is then possible to look through the line from end to end. The copper branches connect to the main leadized header with crosses instead of tees. This type of connection allows cleaning and makes a simple joint.

The new system has been in use for several months and has been opened for inspection several times. The growth of slime has been negligible. Since the building steel is subject to much less vibration than the piping, no difficulties with this source of scale formation are encountered. As a precaution, however, the roof trusses are vacuum cleaned periodically, with remarkably successful results.

Making the Joints

With the copper piping, all fittings and couplings are bored to receive the pipe for which they are made, and provision is made in the fitting for feeding solder directly into the joint. The pipe and fitting are not tinned previous to making the joint proper. In fact it is doubtful if the piping could be assembled that way due to the close fit between the pipe and the bore of the fitting.

In this particular case the tubing was cut into the necessary lengths and the ends were polished with sand paper so as to insure sound joints. The polished portion was then covered with soldering paste to act as a flux to prevent oxidation prior to soldering. The interior of the fittings were prepared in the same way. Whole runs of copper tubing and fittings were then hung on temporary hangers and lined up in place. The joints were then heated with a blow torch and the solder applied.

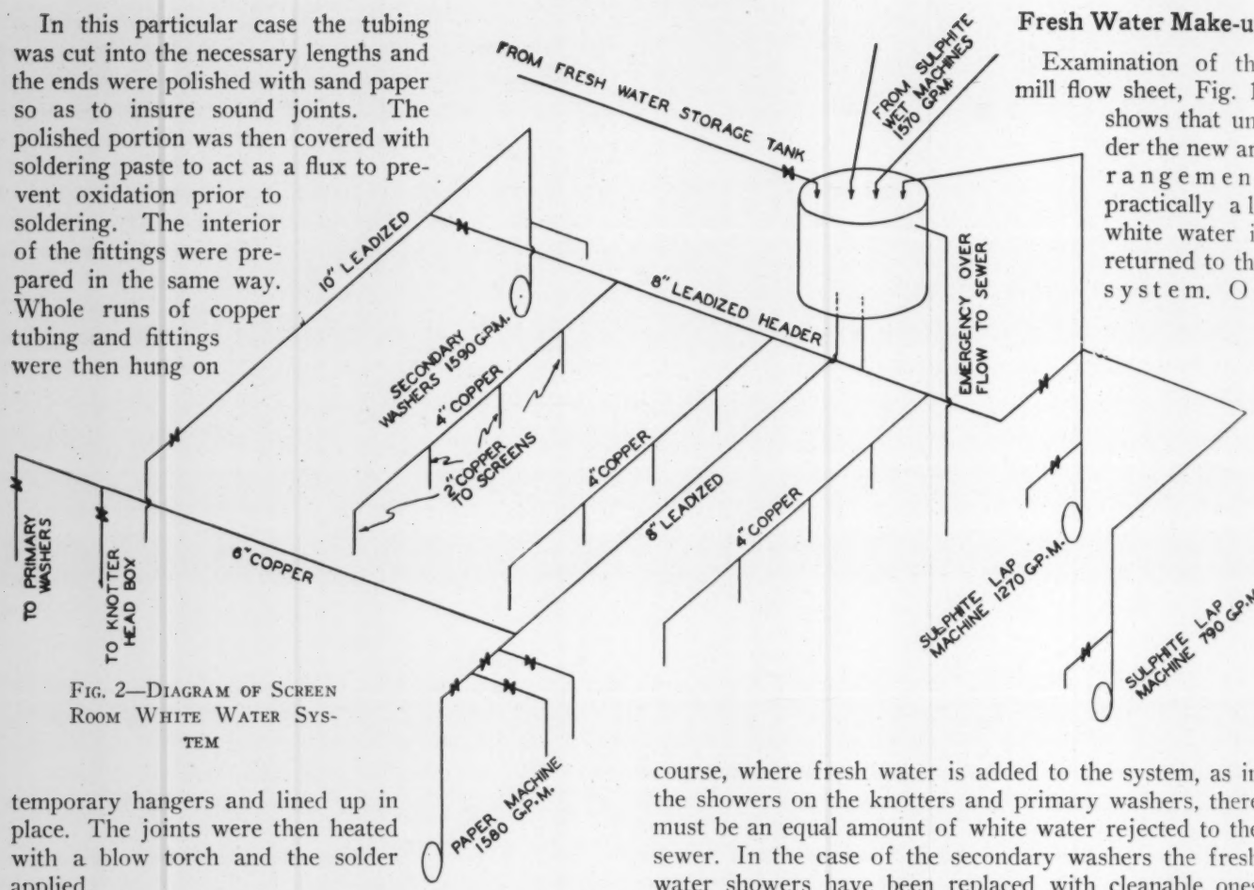


FIG. 2—DIAGRAM OF SCREEN ROOM WHITE WATER SYSTEM

It is estimated that a 6-in. joint was made in fifteen minutes by this method. This includes making one cut, polishing the pipe and fitting, and applying the solder. A 2-in. joint was made in about one-half that time. To date, approximately 300 joints have been made by this method.

Fresh Water Make-up

Examination of the mill flow sheet, Fig. 1, shows that under the new arrangement practically all white water is returned to the system. Of course, where fresh water is added to the system, as in the showers on the knotters and primary washers, there must be an equal amount of white water rejected to the sewer. In the case of the secondary washers the fresh water showers have been replaced with cleanable ones using white water taken directly from the discharge of the removal pump. This change is responsible for a reduction of about 75 g.p.m. in fresh water consumption. It is advisable to add some fresh water to the system, however, to keep it clean, and for that reason the fresh water showers which still remain will be left in service.

Graphic Solution of Chimneys

By H. L. Alt

CHIMNEYS for heating plants seldom receive the consideration which their importance warrants. Frequently, it is found that a power plant chimney has been placed on a heating plant installation and, while the plant will operate, it is neither economical nor correct. On the other hand, chimney tables are not dependable unless they are excessively conservative so as *always* to produce a chimney based on the worst conceivable conditions. This is not economical for installations where the extreme conditions do not, and never can, exist.

As an example, there is no need of putting up a chimney with a height and area capable of caring for a 50 per cent overload on the boilers and of burning No. 3 buckwheat coal under this condition when—as a matter of fact—the entire heating load comes to only 90 per cent of the boiler rating and stokers with forced draft are to be used.

Draft Requirements May Not Determine Height

The increasing popularity of the tall building also has introduced a set of outside conditions which often require a chimney to be carried to a height not necessitated by the actual draft requirements. In such instances the diameter may be reduced correspondingly to the advantage of a lesser first cost and an increase in rentable floor area. For instance, if a 10-story building requires a stack 6 ft. in diameter, a 20-story building might be able to get along with a 5 ft. diameter stack for the same load. Allowing for the enclosing walls, the first chimney would occupy a floor area of 8 ft. x 8 ft. or 64 sq. ft. and the second would occupy only 7 ft. x 7 ft. or 49 sq. ft., saving 15 sq. ft. per floor or 300 sq. ft. in the floor area of a 20-story building. At $2.00 per sq. ft. per year rental this would increase the revenue received from the building about $600.00 per year, an item deserving consideration.

It is evident that, while it is imperative to have a chimney suitable for the required draft and of proper

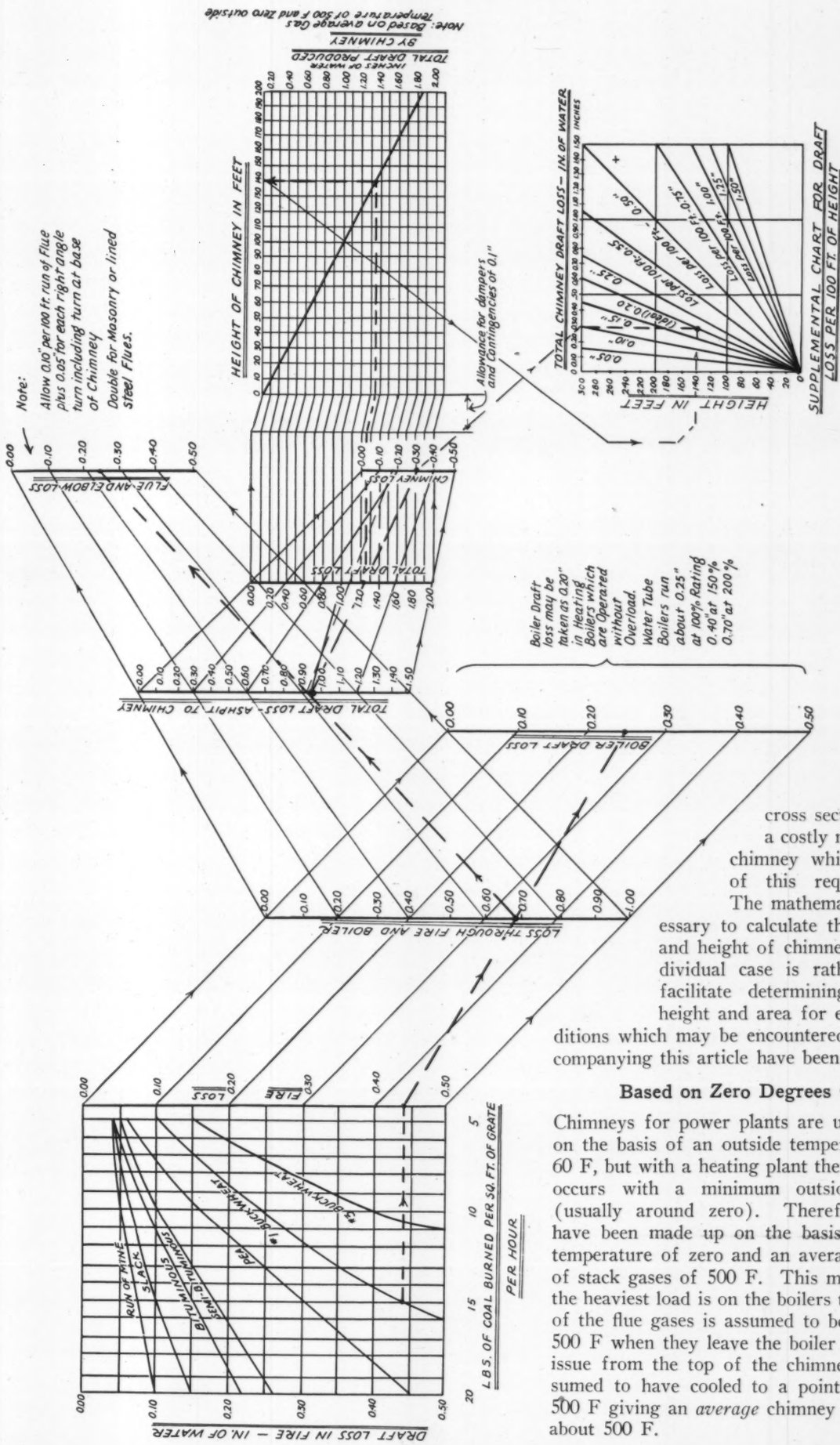

Note: Based on average Gas Temperature of 500 F and Zero outside
BY CHIMNEY
TOTAL DRAFT PRODUCED
INCHES OF WATER
HEIGHT OF CHIMNEY IN FEET
Allowance for dampers and contingencies of 0.1"
TOTAL CHIMNEY DRAFT LOSS – IN. OF WATER
HEIGHT IN FEET
SUPPLEMENTAL CHART FOR DRAFT LOSS PER 100 FT. OF HEIGHT
Note:
Allow 0.10" per 100 ft. run of Flue plus 0.05" for each right angle turn including turn at base of Chimney
Double for Masonry or lined steel Flues.
FLUE AND ELBOW LOSS
CHIMNEY LOSS
TOTAL DRAFT LOSS
Boiler Draft loss may be taken as 0.20" in Heating Boilers which are Operated without Overload. Water Tube Boilers run about 0.25" at 100% Rating 0.40" at 150% 0.70" at 200%
TOTAL DRAFT LOSS – ASHPIT TO CHIMNEY
BOILER DRAFT LOSS
LOSS THROUGH FIRE AND BOILER
LOSS
FIRE
#3 BUCKWHEAT
#1 BUCKWHEAT
PEA
B.T.U. MINIMUMS
RUN OF MINE
SLACK
LBS. OF COAL BURNED PER SQ. FT. OF GRATE PER HOUR
DRAFT LOSS IN FIRE – IN. OF WATER

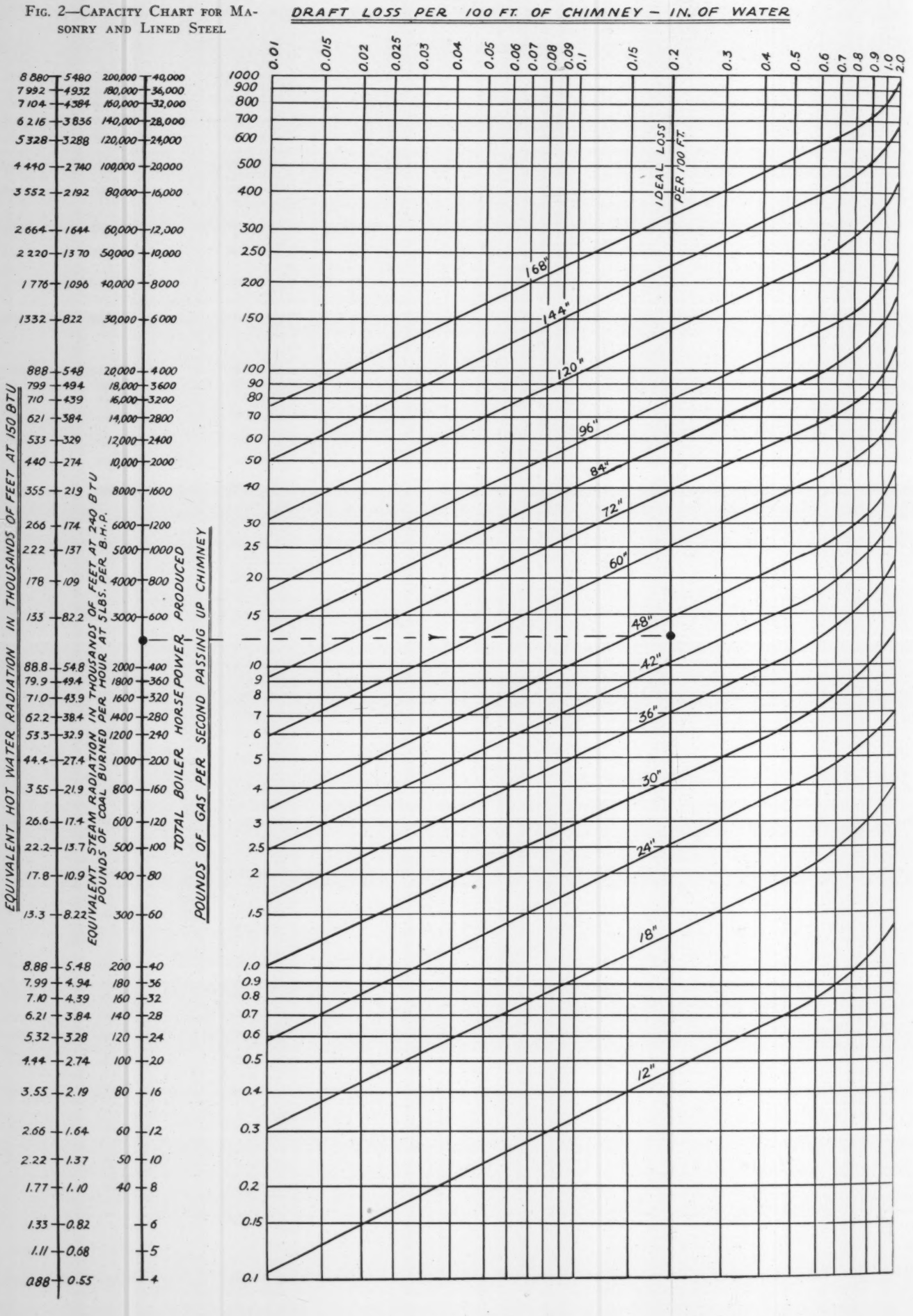
FIG. 1—INTENSITY AND HEIGHT CHART

cross sectional area, it is a costly matter to have a chimney which is in excess of this required capacity. The mathematical work necessary to calculate the required size and height of chimney for every individual case is rather tedious; to facilitate determining the required height and area for each set of conditions which may be encountered the charts accompanying this article have been prepared.

Based on Zero Degrees Outside

Chimneys for power plants are usually designed on the basis of an outside temperature of about 60 F, but with a heating plant the maximum load occurs with a minimum outside temperature (usually around zero). Therefore the charts have been made up on the basis of an outside temperature of zero and an average temperature of stack gases of 500 F. This means that when the heaviest load is on the boilers the temperature of the flue gases is assumed to be slightly *above* 500 F when they leave the boiler and when they issue from the top of the chimney they are assumed to have cooled to a point slightly *below* 500 F giving an *average* chimney temperature of about 500 F.

FIG. 2—CAPACITY CHART FOR MASONRY AND LINED STEEL



Draft Loss in a Chimney

The draft loss in a heating system may be divided into the following items:

Fire, ashpit and grate.
Boiler.
Flue or breeching loss.
Chimney loss.
Damper and miscellaneous.

Use of Intensity and Height Chart

Fire Loss

On the chart shown in Fig. 1 the draft loss is started off with a series of curves indicating the fire, ashpit and grate loss occurring when different grades of coal are burned at various rates of combustion. To use the chart find the pounds of coal burned per sq. ft. of grate per hr. on the scale at the bottom of the diagram on the left hand side and proceed on this line vertically toward the top of the sheet until the curve for the particular kind of coal involved is intersected; then from this intersection proceed directly to the right until the scale marked *fire loss* is encountered where the draft lost through the fire may be read.

Use With Stokers

If stokers with forced draft are to be used, there is no fire loss as far as the chimney is concerned, but it is desirable to maintain a slight negative pressure in the fire box so in such a case 0.1 in. would be found on the *fire loss* scale and the chart reading would begin at 0.1 in. instead of the point opposite the *pounds of coal burned per sq. ft. of grate* and *draft loss in fire* intersection.

Total Draft Loss, Ashpit to Chimney

By laying a ruler on the *fire loss* scale so that the side touches the fire loss found (or the 0.1 in. in the case of stokers with forced draft) and the draft loss occurring in the boiler on the *boiler draft loss* scale it will be found that the side of the rule crosses a third scale midway between the *fire loss* and *boiler draft loss* scale from which third scale may be read the *loss through fire and boiler*. Keeping the side of the rule on the point determined for the loss through boiler and fire, and swinging the rule so that the side falls across the *flue and elbow loss* at the proper point for the breeching in question, from the intermediate scale the *total draft loss—ash pit to chimney* may be read.

Total Chimney Height

Swinging the rule again to cross the *total draft loss—ashpit to chimney* scale at the intersection just located, and the scale for chimney loss at the proper point gives a reading on the *total draft loss scale* of the required inches of draft to meet the conditions assumed. Proceeding directly along the horizontal lines to the right from this intersection, until the diagonal line of *theoretical chimney draft* is intersected, directly above this intersection is read the required height of the chimney. It will be noted in proceeding toward the right to intersect the chimney draft line, that all the lines make a small offset downward and then continue to the right again. This is caused by an allowance made of 0.1 in. for dampers and contingencies.

The total *chimney loss* then must be changed to *draft lost per* 100 *ft. of chimney* which is done by using the small supplemental chart shown in the lower right hand corner of Fig. 1 where by finding the intersection of the *total chimney draft loss* on the scale at the top of the chart and the *height of chimney* on the scale on the left hand side the *draft loss per* 100 *ft.* may be determined by referring to the diagonal lines.

Use of Capacity Charts

Having obtained the allowable draft loss per 100 ft. in the chimney, the charts shown in Figs. 2 and 3 will be considered. The chart shown in Fig. 1 for *intensity* must always be used but the charts shown in Figs. 2 and 3 are for *volume* and are alternate charts; Fig. 2 is the chart for masonry chimneys, and for lined steel chimneys; Fig. 3 is a chart for unlined steel stacks *only*. However, both charts operate in exactly the same manner so that an explanation of Fig. 2 will also indicate how Fig. 3 is used.

Volume of Gases Chimney Must Handle

The volume of gases which a chimney must handle is determined by the maximum boiler load, which may be expressed in several ways:

Maximum boiler horsepower to be produced.
Maximum pounds of coal to be burned per hour.
Maximum pounds of air per second required for combustion.
Number of equivalent sq. ft. of direct steam radiation (at 240 Btu per sq. ft.) supplied.
Number of equivalent sq. ft. of hot water radiation (at 150 Btu per sq. ft.) supplied.

Under average conditions there is a relation[1] between all these items and they may be plotted against each other. This has been done in the scales at the left of Figs. 2 and 3 so that the maximum load when expressed in any of these possible designations may be used to work out the chimney size directly.

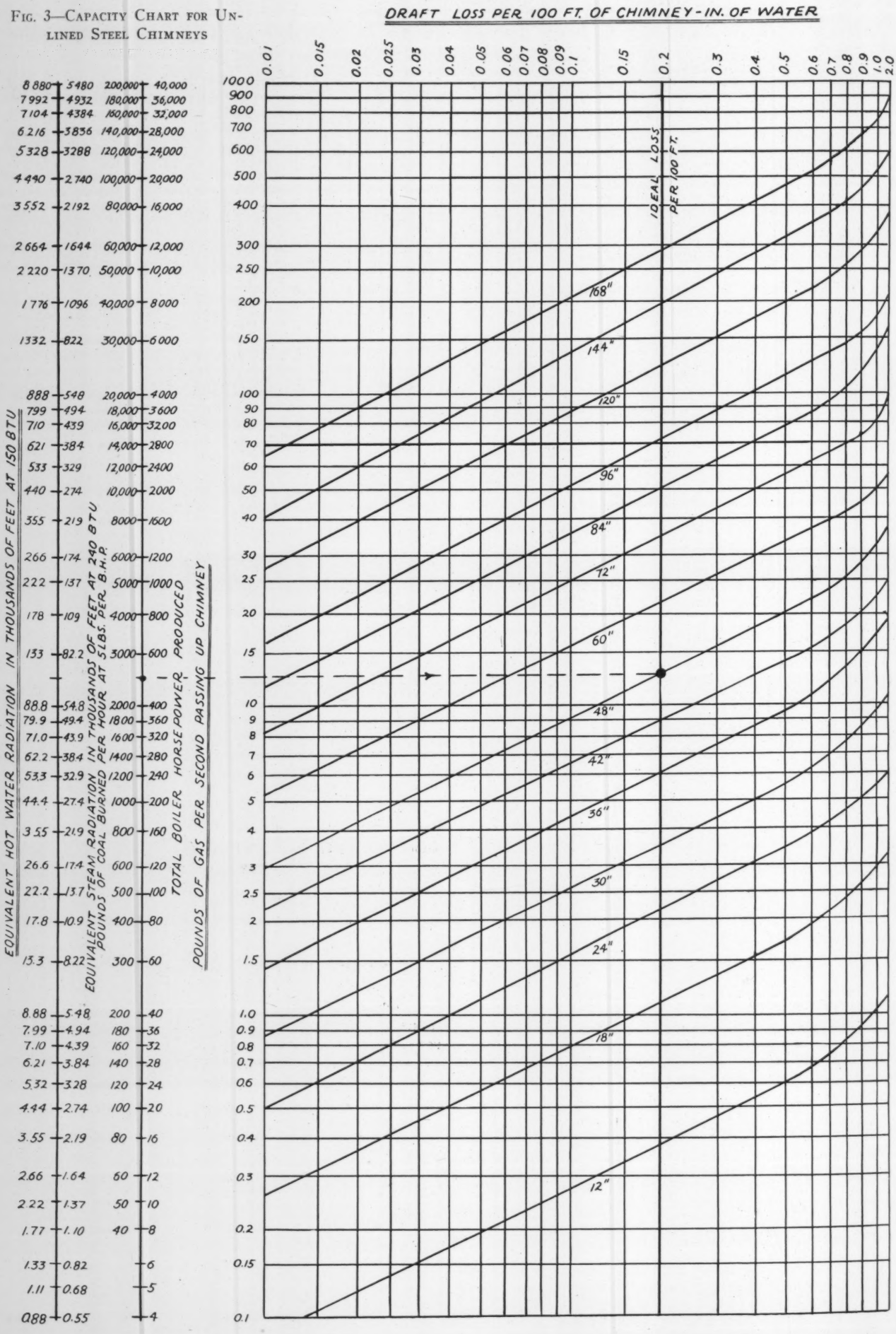
It is quite likely that 15 lb. of air per pound of coal would be sufficient but a small factor of safety has been introduced here (amounting to (18–15) ÷ 15 or 20 per cent) in the same manner that a small allowance was made in working out the draft intensity by allowing 0.1 in. for *dampers and contingencies*.

Using Fig. 2

To use the chart Fig. 2 (assuming a masonry chimney), find the boiler horsepower, the pounds of coal per hour or the load in sq. ft. of steam or hot water radiation, on the proper scale at the left. Then proceed directly toward the right from this point and read the pounds of gas per second from the scale on the left hand edge of the chart. Continue toward the right until the vertical line for the *chimney draft loss per* 100 *ft.* is encountered. The intersection will indicate—by its position in relation to the diagonal lines—the diameter (or the side of a square chimney) required. When such a point falls between two curves the larger diameter should be used although a proportional diameter is allowable if close work is required.

[1] Assumptions:
18 lb. of air required to burn 1 lb. of coal.
1 lb. coal supplies 8,000 Btu to water and steam in boiler.
1 boiler horsepower requires the combustion of 5 lb. coal.

FIG. 3—CAPACITY CHART FOR UNLINED STEEL CHIMNEYS



Having found this intersection and determined the required diameter of the chimney, the height and diameter then are known and, if a chimney loss of about 0.2 in. per 100 ft. has been used, an ideal chimney from an economical standpoint has been determined. For an unlined steel chimney the same procedure would be followed on the chart shown in Fig. 3 for unlined steel chimneys. Should this not meet the conditions of the building the chimney can be modified to do this—provided, of course, that the conditions are not such as to make the chimney perform the impossible. How the charts are handled to modify the ideal relations between height and diameter will be explained by means of a typical case.

Modifying the Ideal Chimney

The Problem: Assume a plant which is to burn 15 lb. of No. 1 buckwheat coal per sq. ft. of grate per hr. with three 250 hp. water tube boilers (one of which is a spare) run at 100 per cent of rating, a steel flue 25 ft. long with four 90 deg. turns between the boiler smoke outlet and the chimney (including the 90-deg. upturn into the bottom of the chimney). What would be the ideal height and diameter of a brick chimney and what would be the required diameter if a steel stack is placed in a 20-story building with 250 ft. from grates to top of stack?

The Method: Locating the 15 lb. per sq. ft. combustion rate on the left hand side of Fig. 1, then proceeding up until this line intersects with No. 1 buckwheat coal and to the right until the scale for draft loss in fire is encountered, it is found that the draft loss in the fire will be 0.44 in. Laying a rule from this point down to the boiler loss of 0.25 in. (for water tube boilers at 100 per cent of rating) it is found that it crosses the scale for combined boiler and fire loss at 0.69 in. The breeching loss for steel breeching 25 ft. long will be ¼ of the loss per 100 ft. or ¼ of 0.1 in. which is 0.025 in.; and four 90-deg. turns will add 4 × 0.05 or 0.20 in., making the total 0.225 in. Laying the rule from 0.69 in. fire and boiler loss to 0.225 in. breeching loss gives 0.915 in. for loss between ashpit and chimney. Then laying the rule on this point and 0.3 in. chimney loss (for trial) the total draft loss is shown to be 1.215 in. and running over to the *draft produced* diagonal line the height (140 feet) is read above the intersection. Reference to the small chart in the lower right hand corner for 140-ft. height and 0.3-in. chimney loss shows 0.2 per 100 ft. as the rate of loss in the chimney.

Now going into Fig. 2 (for masonry chimneys) and running across from 500 boiler horsepower, which is the maximum to be produced, to the vertical line for a stack loss of 0.2 in. per 100 ft. the intersection is found to fall a little under the 48-in. diameter chimney. So a 48 in. × 140 ft. stack would be the ideal size for this load and under the conditions assumed.

But with a 250-ft. height necessary in order to get above the roof of the building it is necessary to go back to Fig. 1 and see what a 250-ft. chimney would produce. On the chart for height of chimney it is determined that a 125-ft. chimney gives a draft of 1.2 in. total so a 250-ft. chimney would give twice as much, or 2.4 in.

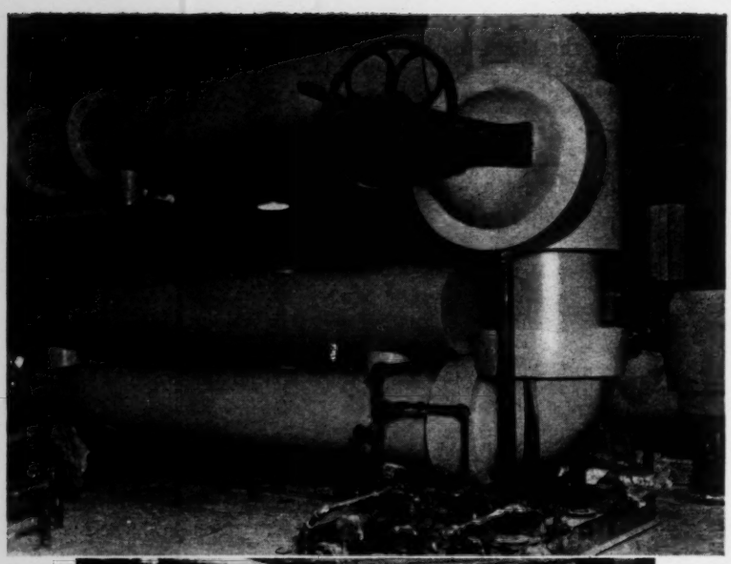
It already has been found that the draft losses up to the stack amount to 0.915 in. plus 0.1 for contingencies, dampers, etc., which leaves for the chimney loss 2.40 — 1.015 or 1.385 in. for a 250-ft. height, which, according to the small chart in the lower right hand corner of Fig. 1 is equal to 0.55 in. per 100 ft.

The Answer: Then going to Fig. 3 (for steel stacks) opposite 500 boiler horsepower and under 0.55 in. loss per hundred feet an intersection is found which falls a little over the 36-in. diameter line, showing that under the conditions of 250-ft. height and a steel stack the same volume of gases will be handled if the stack is made 38 in. instead of the diameter of 48 in. previously developed for a brick chimney 150 ft. high.

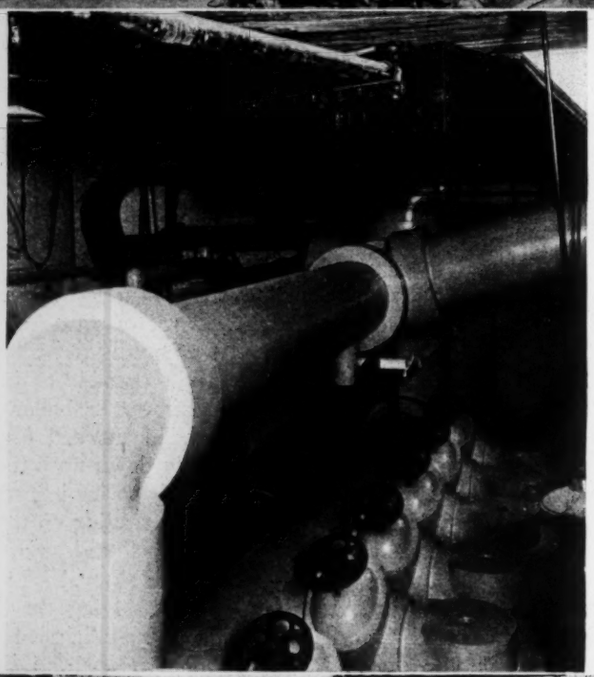
Low Chimneys

There still remains the problem where the conditions do not permit a chimney of a height to produce an ideal condition; i.e. where the friction loss in the chimney must be less than 0.2 in. per 100 feet because the height of stack is insufficient to take care of this amount of loss in addition to the other draft losses between the ashpit and the chimney. With the ideal chimney of 48 in. by 150 ft. it has already been found from Fig. 1 that the draft loss from the ashpit to the chimney is 0.915 in. and with another 0.1 in. for dampers and contingencies this means that the total draft produced by the chimney *after the chimney loss is subtracted* must be at least 1.015 in. or 0.915 in. without the contingency allowance.

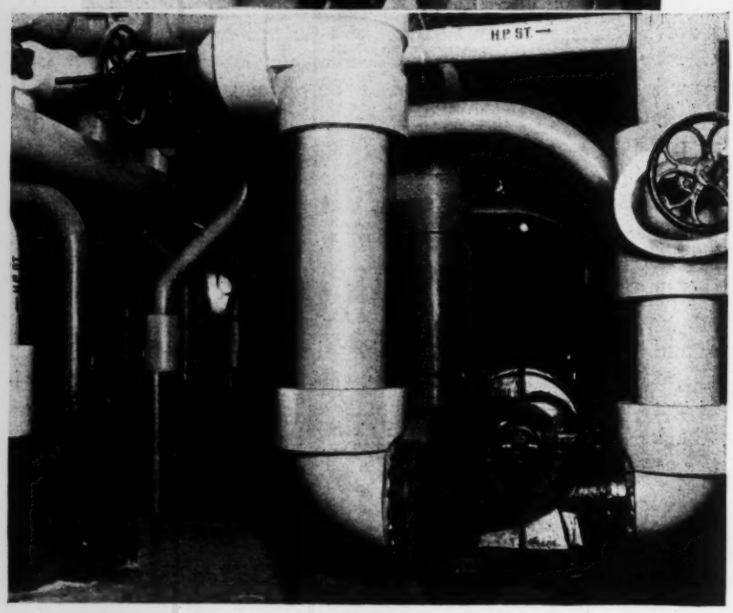
Let it be assumed that it is not possible to obtain a chimney of over 110 ft. height above the boiler grates. Now start from the scale (Fig. 1) marked *height of chimney* at 110 ft., proceed directly down until the draft line is encountered, and then directly to the left until the *total draft loss* line is reached. The reading on this scale shows that the chimney can handle a total loss of 1.0 in. after the contingency allowance is deducted. Laying a rule on this point (1.0 in.) and swinging the rule to 0.915 for all the other draft losses from ashpit to chimney on the *draft lost between the ashpit and chimney* scale, it is found that the rule will indicate on the *chimney loss* scale that the chimney loss must be not over 0.085 in. Applying this 0.085 in. on the small chart against 110 ft. of height indicates a draft loss in the chimney per 100 ft. of 0.077 in. This, then, is the allowable loss per 100 ft. in the chimney and the diameter of the chimney must be made such that this loss will not be exceeded. Then going to Fig. 2 (for masonry stacks) the intersection of 500 horsepower and 0.077 in. loss per 100 ft. is found just below a 60-in. diameter. If, however, the draft produced by any given height of chimney gives a *lesser* amount than the draft lost between the ashpit and the chimney plus 0.1 in. the height manifestly is impractical as no amount of increase in chimney diameter will reduce the chimney loss to zero.



The Grand Central Terminal zone in New York City includes the Graybar, New York Central, Postum, Vanderbilt Concourse, Terminal Postoffice, Park Lexington, Vanderbilt Avenue, Terminal Office, Grand Central Palace and Grand Central Terminal buildings; the Biltmore, Commodore, Roosevelt, Chatham, Barclay, Park Lane, Marguery and Waldorf-Astoria hotels and in addition several apartment houses, all housing 600,000 persons. The annual steam requirements are approximately 1,250,000,000 pounds. The view to the left shows one of the 20-in. steam services from the New York Steam Corporation. The New York Central R. R. is the largest purchaser of central steam service in the world.



Metering is an essential part of the distribution system, and is handled by an experienced staff and operating force. Steam flow meters with variable orifices are used. Regulation of pressure is handled first through service reducing valves, and secondary reductions are handled by the consumers through the installation of secondary reducing valves. Automatic devices control the steam in practically all of the buildings, for economy. This view shows some of the steam metering equipment. Steam is also supplied for high pressure requirements.

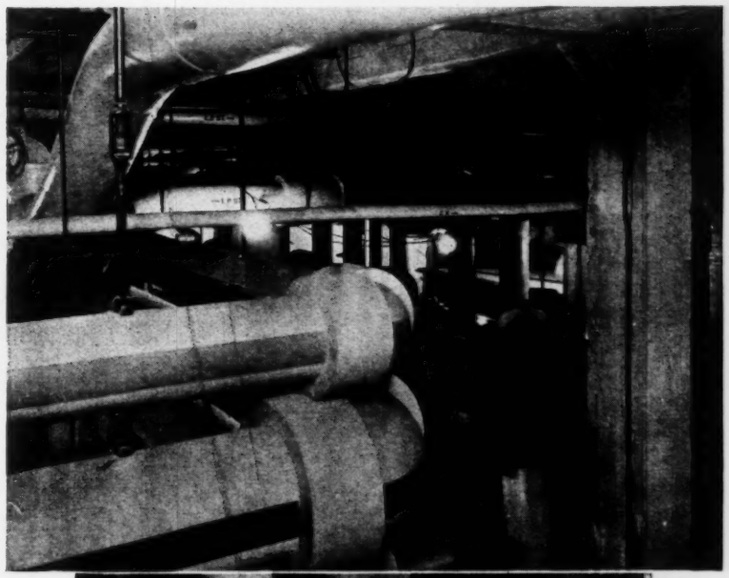


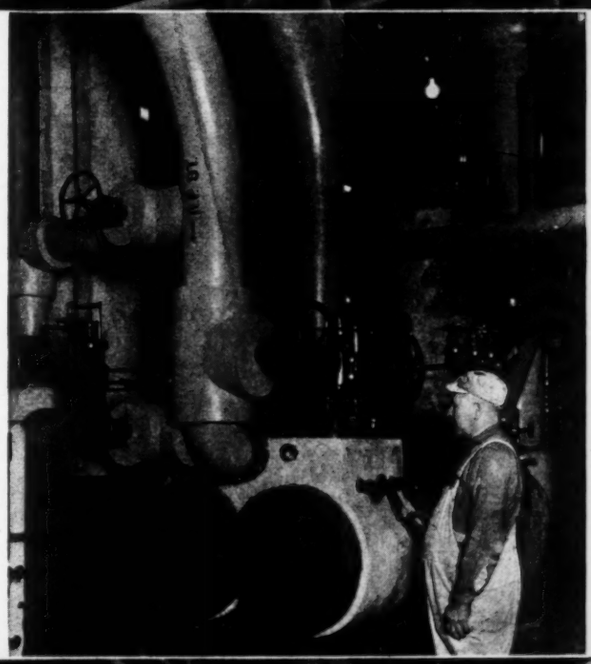
The pump room located in the service plant in the Grand Central Terminal zone. This photograph shows hot water heating circulating pumps on the right. Five large heat exchangers are used, which will raise 6,000 g.p.m. from 140 to 180 F; they are located in a service plant under the Graybar Building. There are three hot water systems—high, low, and intermediate—four 5,000 g.p.m. and one 1,500 g.p.m. centrifugal pumps (steam turbine driven) circulate the hot water through the systems; all are interconnected for continuity of service. Most of the buildings are heated directly with steam, however.

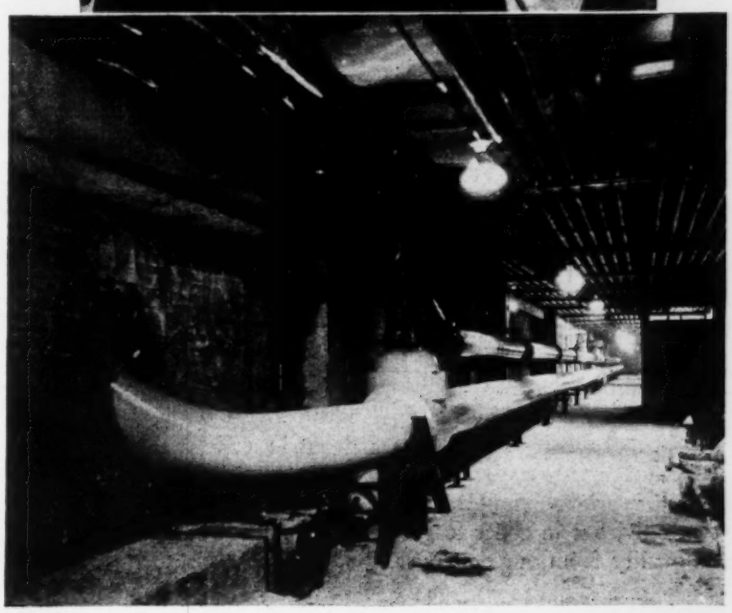
Over a Billion Lb. of Steam Per Year

Here are some interesting views of part of the equipment necessary to supply steam and hot water to the buildings in the Grand Central Terminal group in New York City. These buildings, erected at a total cost approximating $625,000,000, are a veritable city within a city.

The new Waldorf-Astoria Hotel, described in recent issues, is a part of this development.







The top shows headers for high and low pressure steam lines and hot water heating mains under the Graybar Building. The temperature of the water, governed by the outdoor temperature, is controlled by the operator in the service plant, whereby economies are effected. Steam is distributed at approximately 125 pounds pressure, mains being laid out in a complete loop system with ample cross-connections. Sizes of steam distribution lines vary from four to twenty inches.

The fire pump in the service plant is shown in the middle view. The illustration below it shows a twelve-inch steam connection under 43rd street in what is called the trucking subway. A 10-inch water line from the 20,000 gallon tank on top of the Grand Central Terminal building is also shown. The steam line (below) is covered with double standard thickness of 85% magnesia with an outside covering of asbestos cement primed with a sizing coat and two coats of lead and oil; the water line has a one-inch anti-sweat covering.

By W. H. Pierce*

THE MILLION-DOLLAR AMERICAN INSTITUTE OF LAUNDERING, WHERE THE PROBLEMS OF THE INDUSTRY ARE STUDIED

Ventilating and Piping a Modern Laundry

A Discussion of the American Institute Laundry Installations

THAT the laundry industry is a progressive one is evidenced by the great proving and improving institute built at Joliet, Ill. Approximately 2,500 laundryowners of the United States and Canada organized as a national association and pledged enough money to build a million dollar institute where the many problems of the industry could be studied and solved, the facts thus developed to be passed on to the laundryowners for their guidance in the operation of their plants.

The American Institute of Laundering consists of three units: First, a staff of technically trained men versed in textiles, chemistry, accounting, sales and merchandising, and mechanical and industrial engineering; second, a school to which young men and women interested in the laundry business may come for a period of ten months and receive intensive instruction in the five major departments of the business, namely: power plant, production, washroom and textiles, accounting and office administration, and sales and advertising; third, a commercial laundry doing approximately a quarter of a million dollars' worth of laundry per year. It is the third unit that we will discuss in this article.

A laundry business of this size requires approximately 100 people, 70 per cent of whom are girls and women. In designing the institute laundry unit, considerable thought was given to the conditions under which these girls were to work. Our past experience in production tests taught us that the conditions surrounding employes, particularly women, had a great deal to do with the amount and quality of work turned out in a given time. It was our aim to make this institute as nearly a perfect place to work as possible, not from a theoretical standpoint only, but in order to get the maximum production and quality from our employes.

* Engineering Department, Laundryowners National Association of the United States and Canada, Joliet, Ill.

The word "laundry" conveys to most people a picture of an ultra moist, steamy atmosphere. In some cases this is true, as a great deal of water and steam is necessarily used in the washing and processing of work. This condition, however, does not exist in a well designed plant.

Atmospheric Conditions in Laundry

As proof of the statement, a number of tests show a very low relative humidity in the institute laundry. Several tests taken on various days with varying weather conditions produced the following data:

TABLE 1

Location	Wet Bulb	Dry Bulb	Relative Humidity
Outdoors	34	40	51
Starch Department	61	77	40
Flatwork Department	65	85	36
Washroom	64	77	51
Mezzanine (over Washroom)	65	77	52
Mezzanine (Office side)	65	79	46

TABLE 2

Outdoors	42	50	48
Starch Department	63	79	48
Flatwork Department	62.5	83.5	32.5
Washroom	60	76.5	38
Mezzanine (over Washroom)	60	75	43
Mezzanine (Office side)	60	78	36

TABLE 3

Outdoors	34	37	72
Starch Department	64	77	50
Flatwork Department	61	75	46
Rough Dry Department	56	69	43
Washroom	59	71	49
Mezzanine (over Washroom)	60	73	49

The above tables show the relative humidity to be considerably lower on the average than the air out-of-doors.

Amount of Water Evaporated

The expectation of heavily laden air would not be amiss when one considers that each week nearly three-quarters of a million gallons of water are used, 60 per cent of which is heated to 180 deg. and used in machines vented to the atmosphere and naturally giving off vapor.

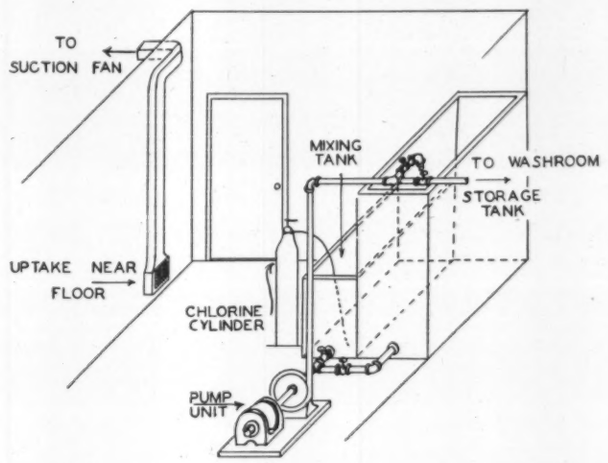


FIG. 1—VENTILATING DUCT IN THE SUPPLY ROOM ARRANGED TO ELIMINATE DANGER FROM CHLORINE GAS

In addition to this, approximately 40,000 pounds of clothes are finished, either by flat ironing, pressing or tumbling. In either case, it means evaporating about one-half pound of water per pound of clothes, or changing 20,000 pounds of water into vapor which passes from the machine to the atmosphere.

The ceiling height of the institute is 28 feet from the floor to the main portion of the ceiling. There are two full length monitors running east and west, which extend another 7 feet above the main portion of the ceiling. Both sides of these monitors are fitted with ventilating sash type windows mounted on an opening and closing mechanism operated from the floor by chains. These may be adjusted to suit the air requirements, much the same as a damper in a stack.

The combination of the ventilation sash in the walls and the monitor construction on the roof form the major part of the ventilating system in this building. These, of course, are supplemented by mechanical means, which will be discussed later.

Special Ventilating Problems

The agent most commonly used for brightening white work and removing stains is a chlorine solution which, when it comes to the washroom, has a strength about the same as the Dakin solution generally used in hospitals for purifying and cleansing wounds. The available chlorine usually does not exceed one per cent. This is used in the proportion of two quarts to each 100 pounds of white clothes. While this solution is diluted a great deal when applied to the washer, it is essential that it be made up in definite proportions, as a mistake in the preparation of such a supply might cause a weakening of the fabrics handled.

To insure the proper strength of this brightening or clarifying agent, it is made up by allowing 100 pounds of liquid chlorine to pass from a steel container into a caustic solution. The entire tank is emptied at one time, as chlorine is a dangerous gas and there is always the chance that a valve may be defective or not properly shut off after being used.

As insurance against this danger, the supply is purchased in a heavy steel drum equipped with a valve. The steel drum is taken into a room used exclusively for making up the clarifying agent and for the preparation of liquid soap. This room is kept closed and locked. It is the job of one of the maintenance men to see that the supply of clarifying agent and soap is maintained. In this way the danger is confined to this one man, who is instructed in the proper operation of equipment and supplies. As a further precaution to insure this man's safety, a special ventilating unit is placed in the room.

Those who had experience with chlorine gas during the late war know that it is heavier than air and will settle to the lower part of a room. The ventilating system, then, must be constructed to remove the gas in case of a leaky valve or broken drum. Fig. 1 shows the construction and location of the ventilating duct in the supply room.

The entire system of pump, pipes and valves to carry the clarifying agent to the washroom is of a special rust-resisting material. Ordinary equipment or piping would be of very short life if used. The cost of the alloy installation is quite large compared to plain piping or pumps, because of its metallurgical and physical make-up. The pipe can not be threaded, but must be cast; joints and unions are made by clamping flanged ends together. The pipe comes in standard 3-foot lengths. Other sizes must be ordered to measurements, which necessitates a very accurate layout on the part of the piping contractor.

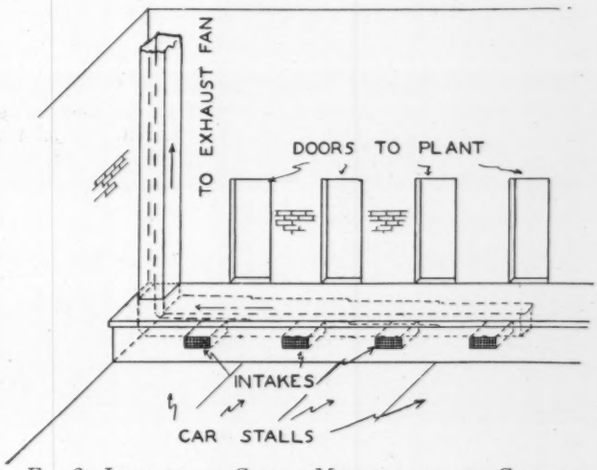


FIG. 2—INTAKES FOR CARBON MONOXIDE IN THE GARAGE

The Garage

The garage is built integral with the laundry unit. It is under the same general roof, but divided from the laundry proper by a brick wall with a number of fire doors through which the routemen load their trucks. There is a loading platform just outside the fire doors up to which the routemen back their trucks.

Danger due to carbon monoxide was eliminated by the installation of a duct passing under the platform and

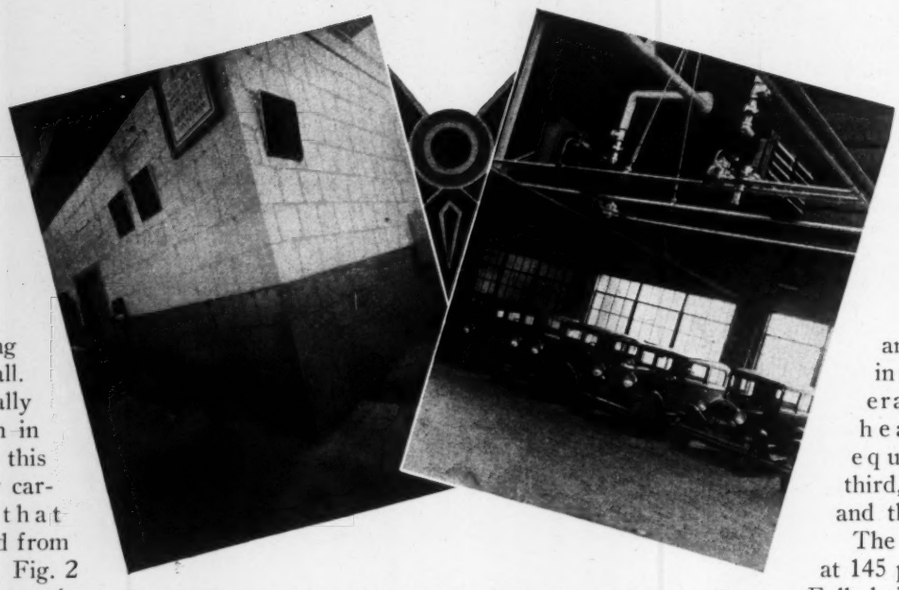


FIG. 3 (RIGHT)—UNIT HEATERS ARE USED IN THE GARAGE. FIG. 4 (LEFT)—A UNIT HEATER IS INSTALLED AT ONE END OF THE LAUNDRY. AIR IS COLLECTED AT THE FLOOR LEVEL AND FORCED OUT THROUGH THREE DUCTS

having an opening at each truck stall. A fan continually creates a suction in the duct and in this way collects any carbon monoxide that might be expelled from an idling motor. Fig. 2 shows the construction and location of the openings, duct and fan which make up the unit.

There was a problem of heating the garage, as the weather sometimes falls below zero and the doors to the garage are open much of the time. It was solved by the installation of two double unit heaters as shown in Fig. 3. There are four single units, two facing each end of the garage. They may be operated as individual units, or all at once, as the weather conditions demand.

Heating the Laundry Proper

The laundry proper is heated chiefly by the heat given off from the processing machinery, such as flatwork ironers, presses, shirt finishing equipment, etc. Forty per cent of the total steam generated in the laundry is used to heat such equipment. A majority of the heat is utilized in evaporating the moisture from the clothes being finished; however, a certain amount is given off by radiation. It is impossible to keep the heating surface in contact with work to be finished at all times. This radiated heat, along with the absence of excess vapor or moisture, accounts for the low relative humidity and the pleasant working conditions.

The only time that heat in addition to that given off by the equipment is required is the first half hour or so in the morning, especially on Monday. As a means of supplying this extra heat, radiators are placed under a number of the ventilating sashes to warm the air passing over them. These radiators are hung on the walls by brackets.

A unit heater installed at one end of the laundry in which very little heated machinery operates also assists materially. Fig. 4 shows the installation. It is comprised of a fan which collects air through a duct from the floor level and forces it through a bank of low pressure steam coils and out through three ducts that tend to distribute the warm air over that portion of the building.

Steam Piping

Steam has three uses in the laundry: First, running an engine, which in turn runs a generator; second, heating finishing equipment; and third, heating water and the building.

The boiler operates at 145 pounds pressure. Full boiler pressure is used on the steam supply to the uniflow engine. In passing through the engine the pressure is reduced from 145 pounds to 2½ pounds. This exhaust steam is then directed into the hot water heater and heats approximately 50 per cent of the required hot water from 55 to 180 F.

About 40 per cent of the heat required is furnished by a heat reclaimer, which is a device for reclaiming the heat from waste water after it has passed through the washer and is on its way to the sewer. This heat reclaimer consists of a bank of copper tubes and a cast header. Incoming water that is to be heated is led through the coils. Warm waste water which averages 125 F is trapped in a pit in which the coils lie. The warm water settles and is cooled off as it nears the bottom, due to heat lost to the cool water in the tubes.

A heat reclaimer will usually give incoming water a rise of from 40 to 60 F, depending upon the time of year and location. Its use results in a fuel saving of from 18 to 23 per cent in this laundry and lowers the boiler load.

The remaining 10 per cent of the heat required for heating water to 180 F is supplied by live steam under thermostatic control at 2½ pounds pressure. The exhaust steam mentioned above includes exhaust from the boiler feed pump, if it is being used, and the small slide valve engine used to operate the stoker.

Steam used for processing work in the flatwork ironers, presses, etc., is reduced from 145 pounds to 105 pounds, which insures dry steam at the machine at 100 pounds pressure. Steam at 100 pounds pressure is safe from the pressure standpoint and also is below the scorching point of cotton and linen. The reduction in pressure from 145 to 105 pounds produces a few degrees of superheat. This small amount of superheat is just about enough to make up for any radiation losses in the supply line.

The boiler room has a 40 foot ceiling, which allows a good rise for a goose neck construction from the boiler to the distributing header. There are three lines taken off the top of the header, an 8-in. line going to the engine

at 145 pounds pressure, a 6-in. line going to the main plant at 105 pounds pressure, and an 8-in. line at 2½ pounds pressure going to the plant heating and hot water system. All steam pipes are covered with 2 in. of 85 per cent magnesia insulation to minimize condensation and heat loss.

Pipe Tunnel Runs Length of Building

Pipes for distributing steam to processing machines are carried through a tunnel 6½ ft. high by 6½ ft. wide running the full length of the building. The pipes are supported along the side of the tunnel, the steam and return pipes on one side, and the hot and cold water pipes on the other. With 6-in. pipes well covered and supported on each side, there is still a 3 ft. clear space down the center of the tunnel for maintenance men to work in.

Supports

Pipes carrying the steam from the main supply pipes run through trenches at right angles to the main tunnel. These trenches are 15 in. deep and 15 in. wide and are covered with steel plates. The pipes rest on supports that keep them several inches above the bottom of the tunnel to insure their not getting wet in case water gets into the tunnel when the floor is scrubbed.

Care was taken that all supply pipes were of ample size to insure each machine getting steam at a minimum of 100 pounds pressure. All low points in supply and distributing lines are trapped to remove condensate as fast as it is formed, thus insuring dry steam at the equipment.

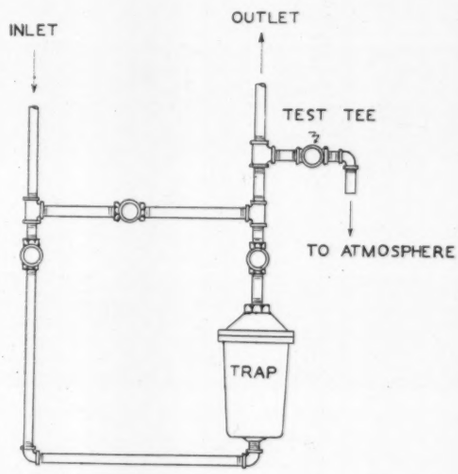


FIG. 5—A BYPASS IS PROVIDED SO THAT THE TRAP MAY BE REPAIRED OR REPLACED WITHOUT INTERRUPTING PRODUCTION

Traps and Return Lines

Clothes are dried and finished by the heat given off when steam condenses. This brings up the problem of removing the condensate from the machine and at the same time retaining the steam. At the institute laundry it is done by placing a steam trap on the drainage end of each machine, or group of machines, depending upon their size.

Both the inverted bucket and float types of trap are used. All large machines, such as flatwork ironers and tumblers, are individually trapped. Some machines, such as the two cylinder flatwork ironer, require two traps for good operation.

Smaller machines such as presses, shirt finishing equipment, hose forms, etc., can be grouped and drained by one trap. It is good practice not to connect more than two machines to one trap, for one machine not properly draining may affect the others in the same group.

The individual trap is placed directly below the outlet from the steam chest, where possible. It is installed with a bypass, as shown in Fig. 5, so that the trap can be removed and either repaired or replaced without interfering with the production of work. A strainer is also installed ahead of each trap to collect any dirt that might come from the steam chest. Such a strainer will save much trouble in the maintenance of a trap.

It is also advisable to have a means of testing a trap without removing it. A simple device for doing this was installed by placing a tee on the discharge side of the trap between the trap and the check valves. By fitting a nipple and a globe valve on the tee it is possible to tell just what the action of the trap is by opening the valve. If the trap is working properly, there will be an intermittent discharge. If the trap is stuck, there will either be no discharge or a steady flow of steam, either one indicating that the trap needs repairing. A swing check valve was installed between each trap and the main return line.

Return lines run through the same trenches that contain the steam lines. These drain to a main return line running just below the steam lines in the main tunnel. The returns or condensate are forced back to a receiver in the boiler room and are then returned to the boiler by means of a float controlled return trap. All return pipes are covered to reduce radiation.

Liquid Soap System

It is easy to use soap as a liquid, and this is the method adopted by the institute laundry. The system includes a unit of two tanks, a pump for circulating the soap through the washroom, and the necessary pipes and valves for its distribution, as shown in Fig. 6.

A definite number of pounds of soap are placed in the small steel tank and the tank filled with water to within 5 in. or 6 in. of the top. The soap is cooked with a steam cooker until it is clear. This cooking and agitating is done by a simple device made up of a four-way cross with a small steam pipe threaded through a plug in one end and extending a few inches past the openings at right angles to the plugged end. The cooker not only supplies the heat necessary for cooking but also acts as an injector or agitator. Some form of an alkali, such as soda ash or modified soda, is used with the soap to increase its detergent action. An alkali of this sort should not be boiled; therefore, it is put into solution in a small tank located over the smaller of the two soap tanks. When the soap in the small tank is in a clear solution, the steam is turned off and the alkali, which has been put into solution in the small container, is turned into the soap and mixed with a paddle.

This soap and alkali solution is then ready for use with the exception that it is in a concentrated form. It has been found advisable (from an economy standpoint) to use it in a weaker solution. The solution is, therefore, pumped from the small soap tank into the large one and hot water added at the same time. The soap is diluted until the large tank is practically full. The valves are then manipulated so that the pump takes diluted soap from the large tank and circulates it to the washers in the washroom.

The 2-in. distributing pipe circles the entire washroom. A supply pipe runs from the main line to a supply bowl at each machine. The amount of soap going to each machine is controlled by a self-closing whistle type valve. That is, the washman must hold the valve open during the time the soap is pouring into the supply bowl. When he has enough, he releases the valve lever and the valve automatically closes.

The main distributing pipe leads back to the large supply tank. A valve is placed on this return line and throttled until a pressure of about 7 pounds has been built up in the 2-in. supply line. This insures fast filling of the supply bowl. The entire pipe system is 230 feet long and has a rise of 13 feet. Under these conditions a pump capable of supplying soap at a rate of 35 gallons per minute against a 75 foot head is used.

A low pressure steam line is run to each of the two soap tanks. Hot water is run to the two tanks and also has a connection to the pipe on the suction side of the pump.

About 15 minutes before closing time each night, the control valve from the soap tank is closed and the hot water supply valve is opened. Hot water is pumped through the entire pipe system and forces the soap back into the storage tank for the night. The hot water remaining in the pipe is drained and the pipe is then clean and free of all material and ready for use next morning.

Some of the first liquid soap installations called for a ¼-in. steam pipe on the inside of the 2-in. supply line to keep the soap from hardening. By using a dilute solution and constantly circulating it, this is not needed.

One very important item to remember in piping steam to a liquid soap system is the installation of a tee with a swing check valve opening to the atmosphere. This is done in case pressure goes down in the boiler at night, thus creating a vacuum which will tend to pull liquid soap from the supply tank back into the boiler. The swing check valve will be closed when the steam pressure is exerted on it, but will open if a vacuum is set up. Air will then be pulled into the boiler rather than liquid soap. Several instances of soap being pulled into the boiler over night have been reported in installations which did not provide for this emergency. An overflow pipe leading from near the top of the large tank into the smaller one should also be installed.

Water System

The fact that nearly three quarters of a million gallons of soft water are used per week in this plant would suggest that an adequate water system would have to be used.

This water comes from a private well 1,603 feet deep and contains from 18 to 20 grains of hardness. This hardness, of course, must be removed.

Two methods are used to remove the hardness from the water. The water is pumped from the well at a rate of 450 gallons per minute. It goes directly to a 70,000 gallon steel tank where it is mixed with lime to precipitate part of the hardness. Alum is injected into the mixture to cause the lime and hardness to settle to the bottom where it can be drawn off. This treatment removes all but seven grains of hardness.

The water is then pumped through a sand filter and then a softener where the remaining hardness is removed. This system has a capacity of 85,000 gallons per regeneration. The water passes from the softener to a 100,000 gallon storage tank.

The lower half of the storage tank is used exclusively for fire protection, as the plant is outside the city limits and city water is not available. The supply pipe leading from the storage tank to the laundry extends halfway up through the tank. If the tank were half empty, no water would be available for washing. The supply pipe for the sprinkler system is taken off the bottom of the tank so there is always at least 50,000 gallons of water available for fire protection and 100,000 gallons, if the tank is full.

The supply for laundry use is taken from the storage

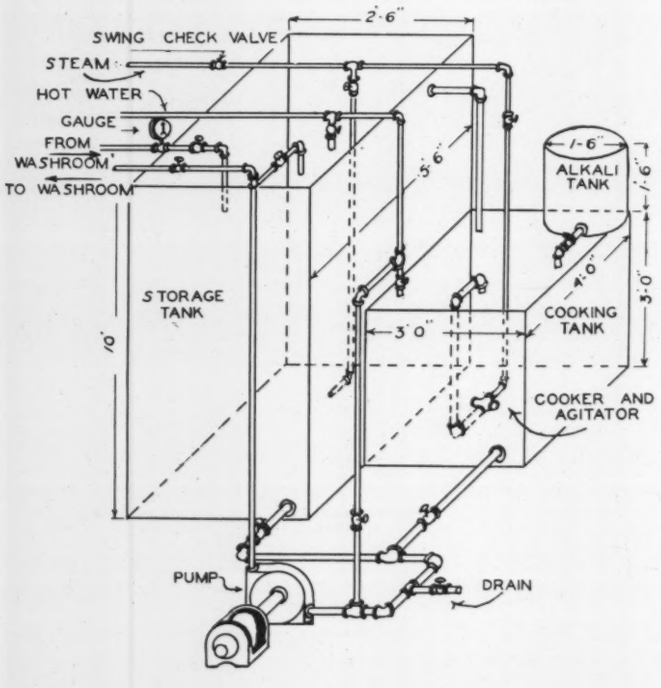
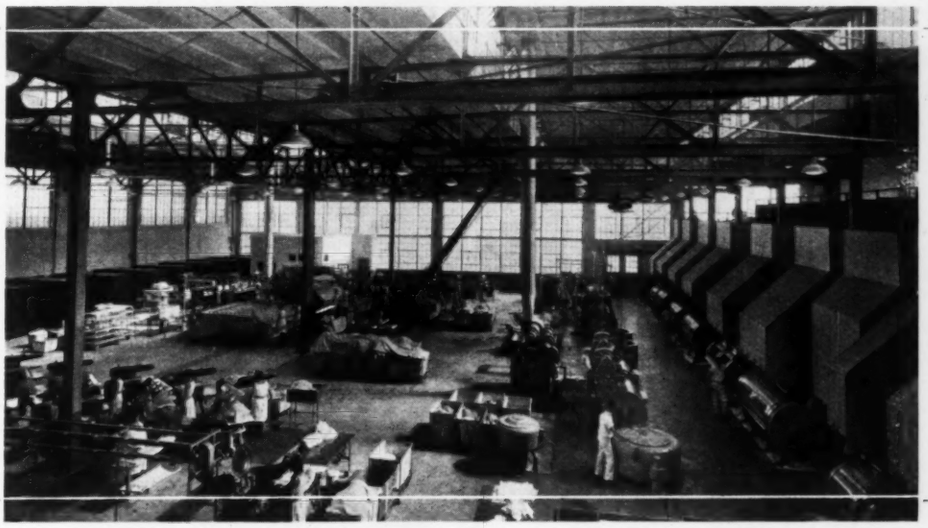


FIG. 6—A DIAGRAM OF THE LIQUID SOAP SYSTEM



tank in two 6-in. lines. One of them goes directly to the washroom as a cold water supply, while the other goes to the bottom of the hot water accumulator, from there to the heat reclaimer, then to the hot water heating unit, and then to the top of the hot water accumulator. The 6-in. line continues from the hot water accumulator or storage tank to the washroom.

Both hot and cold water pipes are taken through the tunnel. They leave the tunnel just back of the washer line. Both hot and cold water are distributed to the washers through 4-in. pipes. All leads to washers are 2½-in. pipes.

An average sized washer uses from 1,000 to 1,500 gallons per load. The water is changed eleven times. One and a half minutes lost at each filling for a day would mean a time loss equivalent to one complete washing, or a loss of about 16 per cent in the efficiency of the machine. Water should be delivered to the washers in adequate pipes and at a pressure of not less than 35 pounds and not over 60 pounds, if possible. This upper limit is given because excess pressure is detrimental to some automatic water valves.

All water pipes, both hot and cold, the hot water system, the filter and softener are covered wtih insulating material. The hot pipes are covered to reduce radiation, and the cold pipes and cylinders to eliminate sweating.

Flow Through Valves

By M. T. Martin*

MAY the writer remark upon an apparently closed correspondence[1] in the May, 1930, issue regarding velocity through valves? The writer concurs in the opinion that the velocity of the steam (or any gas) through a port or orifice is a function of the initial and critical pressure (or else the final pressure where it is above the critical) and therefore is not controllable where the initial and final pressure are fixed. The velocity of 53,000 ft. per min. mentioned in the correspondence is shown calculated as the velocity at the entrance to the seat where the specific volume is that of the initial pressure, although it is mentioned as the velocity through the port.

The writer submits a different method of determining the velocity through the port based on the assumption that in the expansion of a gas through a port the velocity acquired is that which is equivalent to the energy released through adiabatic expansion and can therefore be found by the application of thermo-dynamic formulas for such expansion or more easily by steam diagrams such as Ellenwoods charts. Inspection of these formulas shows that the critical pressure (that secondary pressure beyond which any decrease will not increase the flow) varies with the adiabatic exponent of the gas involved. Assuming the generally accepted exponent of 1.135 for saturated steam this critical pressure is 57.7 per cent of the absolute initial pressure. (For superheated steam in the average field the exponent usually used is 1.3, giving a critical pressure of 54.6 per cent of the initial.)

Where the expansion, as in the example cited, is from 100 lb. gage to 50 lb. gage or less, the critical pressure,

* Webster Tallmadge & Co., Inc., New York City.

[1] Sizing Pressure Regulators, p. 416, May, 1930, Heating, Piping and Air Conditioning.

"Open For Discussion"

51.6 lb. gage, will be close behind the minimum constriction of the port at which point the specific volume will be that of steam having expanded adiabatically to the critical pressure, and the velocity at that point will be the velocity given to the steam by the energy available from such expansion. Were there no restriction coefficient to be applied the specific volume at this point will be 6.3 cu. ft. per lb. and the available energy 43.5 Btu. This energy will give to the steam a velocity of 88,500 ft. per min. which is the velocity through the port, giving a flow of 5,850 lb. per sq. in. per hr., a difference (partial slide rule calculation) of some 1¼ per cent and 2½ per cent respectively from the value to be obtained by the more empirical flow formulas of Napier and Grashof. Where the secondary pressure is lower than 51.6 lb. gage, the expansion from the critical takes place after the steam emerges from the port into a larger area. Where the secondary pressure is above the critical there is of course no further expansion behind the port.

While the above is the theoretical flow through a perfectly formed port the actual flow will be much less due to eddies around sharp corners, changes of direction of flow and due to the effect of the steam passing from a relatively large chamber through the narrow port forming a sort of vena contracta, decreasing the effective area of the port and requiring the introduction of a coefficient due to this contraction. This coefficient depends upon the formation of the valve body, disc, and seat, the smoother the approach the higher being the coefficient. This coefficient is not necessarily a constant for all positions of the disc, though usually it is approximately so. In the average globe type of valve this coefficient is usually 0.4 to 0.7.

Trunk Ducts for Warm Air Heating

By O. W. Kothe*

IN COMMENTING on the paper on trunk ducts for fan blast warm air heating,[1] I wish to say the material covered is something for many heating folks to think about. Some of the items covered are quite novel and the general scheme shows a revolution that has been taking place in the indirect heating field.

The author is presenting a timely subject—the application of direct fired heaters to large indirect heating systems. The author's drawings show the courage warm air men are adopting—all of which is an outgrowth of the scientific research carried on at the University of Illinois in past years combined with what has been developed in the indirect field.

With the direct fired plants every convenience and facility can be included that is felt so necessary under modern operation, such as:

1. Air washers for cleaning and humidification.
2. Air filters with special humidification.
3. Ozonation to revitalize the air.
4. Centralized automatic control for the entire system.
5. Individual duct or trunk line systems.
6. Noiseless operation.
7. Simplicity of and economy in operation.
8. They can be used in conjunction with air cooling plants for the summer months.

The paper features several novel ideas. One, in my opinion, is the deep extended plenum chamber shown in Fig. 2 of that article, when used for both heated air and tempering purposes. However, I question its satisfactory operation as a general practice to follow, for the reason that any fan blast always establishes more or less turbulence.

I remember nearly 25 years ago in Butte, Mont., while overhauling the heating system in a school, some tools were left in the tunnel. Monday morning I went for them but the fire was up, and unknown to the engineer I crawled in after my precious hand tools—the air streaming past me. Today I can still feel that sensation of the oddness of the currents—which I call turbulence. Such a condition can not make a difference of only a few degrees in temperature between the top and bottom of the deep tunnel.

If such a slight difference is sufficient and the tendency is in that direction—then I suppose it achieves its purpose. In the past, the general opinion has been to use either outside air, or recirculated air for tempering, which seldom is over 60 degrees. In such a case a relatively small amount of cooler air mixed with a larger body of heated air, will reduce the mixture to a more agreeable temperature. However, if the practice is changing, and it is found that to use air only a few degrees difference in temperature for tempering purposes is better, I am glad to know it. I am sure other such experiments would be interesting.

Another novel feature is the double duct indicated in Fig. 4. This, to my way of thinking, has greater possibilities than the large deep plenum chamber. Compactness is one of the modern requisites, and this double-duct scheme shows a tendency in the right direction. Certain refinements will no doubt be made; but the idea is good, and in my opinion is applicable to all such indirect duct systems.

Judging from the drawings, I gather all flues are bricked and plastered. Some explanation ought to be given for this. An ordinary brick lined flue is very wasteful of air; surfaces are so rough that the full quota of air is seldom delivered. Men have told me of fans installed over bricked in chases and operating at full force barely making a perceptible flow beyond four or five stories, when the intention was to ventilate eighteen stories.

Rough plaster flues also set up an undue frictional resistance and are also good dust catchers. Only smooth, hard finished plastered flues are comparable with galvanized metal ducts.

In speaking of fans operating continuously and intermittently, I believe a distinction should be made in the design of the duct system. The system with a fan operating continuously is designed identical to all large indirect systems. But where a fan runs only an hour or two at a time, with operation several hours in between as a gravity system—that job should receive attention comparable to a gravity installation. Such a system would really be based on the standard code and the fan is an auxiliary accelerator at desired periods.

Mr. Overton's paper is well worth several readings.

* Consulting engineer, St. Louis Technical Institute, St. Louis, Mo.

[1] *Trunk Ducts for the Fan Blast System of Warm Air Heating*, by Platte Overton. HEATING, PIPING AND AIR CONDITIONING, November, 1930, p. 923.

ELECTRIC BOILERS FLATTEN HYDRO SYSTEM LOAD CURVE —STEAM USED FOR DISTRICT HEATING

Some Reports on Research

All reports on research do not come from laboratories established for the purpose of extending our knowledge. The day-to-day experiences of the consulting engineer, the operating engineer, and the contractor add to our engineering information.

HYDRO Central Heat properties, municipally-owned Winnipeg central heating company, is now valued at $1,185,000. It will soon receive the $450,000 addition voted in 1930. Off-peak electric power will come from the $7,000,000 power development at Slave Falls, Man.

Privately owned plants are to be expanded by the addition of another $1,000,000 plant. This will carry heat to dwelling houses in suburban districts. Nearly one thousand homes now purchase central heat, but the aggregate sale of heat to dwelling houses is much less than steam supplied by the municipally-owned hydro heating service to downtown districts.

Plant Supplies Downtown Districts

"Municipal service through Hydro Central Heat is limited strictly to business uses," states J. G. Glassco, manager of the system. "Part of our territory contains a residential section, but we have not explored this channel, which we consider would be unremunerative at our average present rate of $1.06 per 1,000 lb. of steam.

"Winnipeg Hydro Central Heat was the first munipally owned central heating system in Canada. The plant, put into operation in October, 1924, now operates approximately 40,000 lineal feet of distribution mains connected with some 725,183 sq. ft. of radiation. Steam customers numbered 220 in 1930; office buildings, restaurants, churches, schools, theaters, fire houses, police stations, are included in the varied service.

"Central heating was first undertaken as an economical measure to provide means of utilizing an electrical standby plant. Electrically heated boilers, which were set in addition to the usual fuel-fired apparatus in order to provide an outlet for off-peak energy, afford some useful comparisons in steam production costs.

Plant Divided in Four Sections

"Our plant is divided into four main sections. One is devoted to the handling and preparation of coal for the fuel-fed boilers; a second section is given over to the boiler room, the third to turbine room, and the fourth serves the dual purpose of being the control gallery for the turbine room and at the same time it is substation No. 6 of the hydro-electric system.

Four Water Tube Boilers Used

"The boiler room is equipped with four water tube boilers, fuel-fed, with pulverized coal, and three electric boilers. The water tube boilers have one steam and one mud drum. The rated capacity of each boiler is 1,100 h.p., and each is capable of evaporating a peak of 110,000 lb. of water per hr., gage pressure 250 lb. per sq. in. Steam from three of the boilers is delivered at 150 F superheat.

"Steam from the fourth boiler is not superheated and is used exclusively for central heating. All combustion chambers are of hollow wall construction and each is fitted with a horizontal water screen near the base of the chambers, to prevent fusion of the ash. Combustion chambers are hand-raked, and ash removal from the receiving trench in the floor is done by means of a steam vacuum jet. Bridge walls of boilers 1, 2, and 3 are air cooled; but the water screen in boiler 4 turns up in the rear to water-cool the bridge. Boilers 1 and 2 have induced draft fans for aid in carrying heavy loads or overcoming any temporarily unfavorable draft condition.

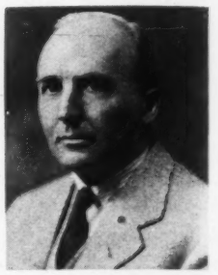
"Pulverized coal is used in the fuel-fired boilers. The bituminous coal used has a proximate analysis of moisture, 1.5; volatile, 23.5; fixed carbon, 60.0, and ash, 15.0 per cent. The calorific value is 12,800 Btu per lb.

"Each of the three electric boilers has a capacity of 7,500 kw. and is capable of producing approximately 21,000 lb of steam per hr. at 250 lb. Electrical energy is supplied at 4,800 volts, 3 phase, 60 cycles.

Electrical Boilers Flatten Load Curve

"The operation of the electric boilers is interesting. They serve primarily to flatten out the load curve of the hydro-electric system by utilizing off-peak energy. The production of steam for use in the central heating system was a secondary matter.

"During 1929 some 90,000 kw.-hr. of off-peak energy were accounted for by the boilers, and increased the hydro-electric annual load factor from 51.1 to 63.2 per cent. Low water conditions in the Winnipeg river during 1930 lowered this figure last year. Nevertheless, the 40,000 kw.-hr. of off-peak energy during 1930 boosted the annual load factor from 53.4 to 56.3 per cent.



J. G. Glassco, Manager, Hydro-Electric System, Winnipeg

Steam Produced

"The steam production by electric boilers resulted in a lower generation cost than fuel-fired boilers. During 1930 the electric boilers produced 119,659,560 lb. of steam, resulting in 2.8 lb. of steam per kw.-hr. The coal-fired boilers produced 306,631,845 lb. of steam, with 8.5 lb. of steam per lb. of coal.

"The peak load of steam per hour was 235,000 pounds, and 410 pounds of steam were sold for the season per square foot of radiation served. The mean temperature for the season was 27.76 F. The steam production figures named include steam supplied to turbines and to plant auxiliaries.

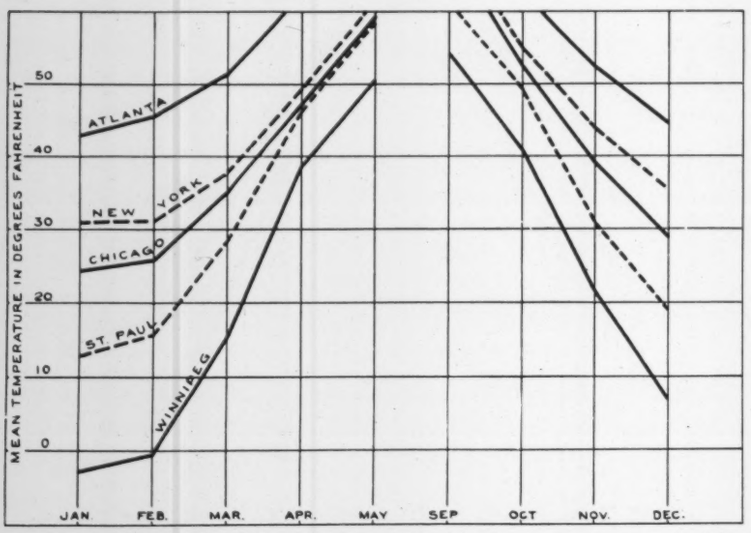
Heating Steam

"The steam for central heat from the boilers feeds into a common loop header from which it passes in turn through a desuperheater and pressure reducer (one set of each at each end of loop) and then into the low pressure header. The steam goes from the low pressure header through separators where all surplus water is drained from it, and thence into the distributing header at 55 to 75 lb. per sq. in. (depending upon the load conditions) with 0 to 35 F superheat, depending upon which of the boilers are carrying the load.

The Steam Mains

"Two 14-in. mains carry the steam from the plant to the outside distribution network. Main line piping is carried in underground concrete trenches, tile-drained to keep them dry. The grade of from 6 to 4 in. per 100 ft. is departed from only when necessary in order to pass obstructions. This grade is sufficient to carry the condensate to sewers. There are no return pipes to the plant.

"Slip expansion joints are used on all lines except the individual service lines. These are placed at intervals of 300 and 250 ft. on 14 and 12-in. trunk lines; and 230, 200, and 150 ft. on 10, 8 and 6-in. lines, respectively. Knuckle joints, placed as close as possible to main line connections, take care of the expansion requirements of the individual service lines.



Seasonal Temperature Chart for Winnipeg and Other Cities which Utilize Central Heat

Rates for Steam

"Steam consumption is measured by metering the condensate in each building served. An economizer installed in front of the meter insures that all steam is condensed.

"The rates for steam are based upon consumption rates per month. The charges vary from $1.40 per 1,000 lb. for the first 25,000 lb. per month to $.90 per 1,000 lb. for all steam over and above 500,000 lb. per month. The average rate received is $1.06 per 1,000 lb.

—*S. P. M.*

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

1895

JULY 1931

Material in this Section prepared by Publication Committee: W. A. Rowe, *Chairman;* E. O. Eastwood, and F. D. Mensing A. V. Hutchinson, Manager of Publications

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Air Infiltration Through Double-Hung Wood Windows

By G. L. Larson[1] (*MEMBER*), D. W. Nelson[2] (*MEMBER*) and R. W. Kubasta[3] (*NON-MEMBER*), Madison, Wis.

This paper is the result of research conducted at the University of Wisconsin in cooperation with the A. S. H. V. E. Research Laboratory

AIR infiltration through various types of building constructions has been the subject of research carried on at the University of Wisconsin for the past several years in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The previously reported work has been on walls of brick and wood frame constructions.[4] The primary purpose of the investigation reported in the present paper was to aid in the establishment of figures for both plain and weather-stripped windows of the double-hung wood type for use in the calculation of heat losses from buildings. As secondary aims, it was desired to study the variation to be expected from one window to another for various cracks and clearances, for locked and not-locked conditions, and for several representative weatherstrips as applied to plain windows showing both low and high resistance to air leakage.

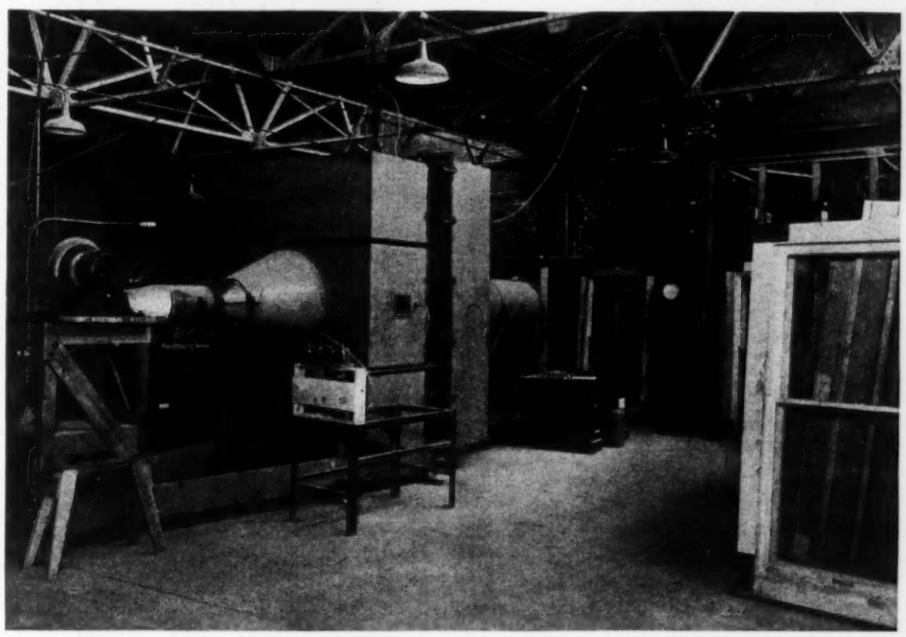

FIG. 1—ARRANGEMENT OF TEST EQUIPMENT

Program and Windows Used

For the investigation, fourteen windows having a sash opening of 3 ft 0 in. by 6 ft 0 in. were purchased. The thickness of the sash was 1⅜ in. in all cases, and for convenience in testing, box frames were used. These differed from standard masonry frames in the additional use of box construction at the head and sill and the use of steel corner plates to stiffen the frames for test purposes. Some of these windows are shown in Figs. 1 and 2.

Five of the windows were purchased a year in advance of the actual time of starting the test, which was in the late Summer of 1930, and from a mill from which windows of poor workmanship and material had previously been secured. These were intentionally specified to be of a poor grade. The other nine of the fourteen were ordered from a mill from which windows of a good grade had been previously secured and were specified to be of the best material and workmanship throughout. These were delivered three months before the testing started. From the time of delivery, all windows were kept in the laboratory to become seasoned before adjusting cracks and clearances or testing.

Six months after starting the tests, the fourteen windows were weatherstripped by local representatives of three weatherstripping manufacturers. Each representative applied his weatherstrip to three of the windows termed to be of a good grade, and to two of the windows considered to be of a poor grade except that one make of weatherstrip was applied to only one window of poor grade since there were only five of this grade. For

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[4] See Air Infiltration through Various Types of Brick Wall Construction by G. L. Larson, C. Braatz and D. W. Nelson (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929) and Air Infiltration through Various Types of Wood Frame Construction by G. L. Larson, D. W. Nelson and C. Braatz (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930, when published).
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

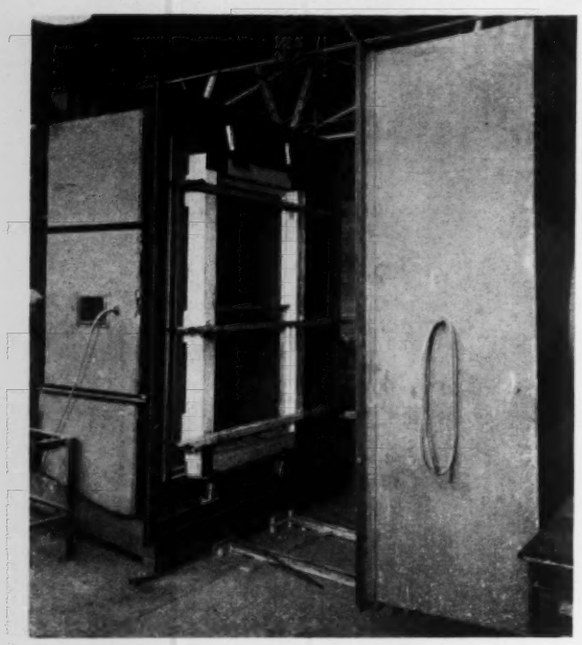

FIG. 2—WINDOW IN POSITION FOR TEST

the purposes of this paper, the windows were designated by letters that follow in sequence for each of the three weatherstrips represented in the tests. Table 1 explains these designations and the symbols that were used on the curve sheets included in this paper.

Test Equipment Used

Fig. 1 shows a general view of the test equipment arranged for window testing. The method of clamping a window in place is shown in Fig. 2. The partition to which the window is clamped divides the machine into a pressure chamber and a collecting chamber. The leakage through the partition is maintained at a low value and the joint between the plane outside face of the window frame and the partition is sealed by the compression of a sponge rubber gasket strip. Distance blocks are located at each clamping location to insure the even compression on the gasket and the maintaining of the window in a plane undistorted condition.

Artificial wind pressure is exerted by a small motor-driven blower against the outside of the window which forms one part of the enclosure for the pressure chamber. The pressure drop through the window as indicated by an inclined draft gage connected to the two chambers is adjusted by means of dampers on the inlet and discharge sides of the blower. Air that passes through the partition wall and window into the collecting chamber is measured as it escapes into the atmosphere by thin-plate, square-edged orifices. The drop in pressure through the orifice is measured on a Wahlen gage. Access doors are provided in each of the two chambers so as to be able to open, close, lock or seal the window during the progress of a test without disturbing the rubber seal between the two halves of the machine.

Test Procedure

Each time a window was placed in the machine for testing, four runs were made. Two runs were made with the window closed but not locked. This was done to obtain a better average value than from one test. Before each of these tests, both sash were raised and lowered several times and the closure made with ordinary effort as it would be in actual building occupancy. Access to the inside of the window was secured through the bolted and gasketed door in the collecting chamber. Since locking places the sash in a definite position, only one run was made at this condition. The air leakage observed at the orifice in the aforementioned runs was the sash perimeter leakage plus a certain amount of leakage through the window frame joints, through the rubber seal joint between the window frame and the machine partition, and leakage occurring through whatever minute cracks existed in the machine partition itself. The latter leakage was kept at a low value by sealing all visible cracks with calking compound and by the application of asphalt paint.

Each time a window was placed in the machine for test, a fourth run was made in which the entire sash perimeter was sealed on the outside of the window to prevent all leakage of air at the joints between the sash and the frame at the head, sides or jambs, meeting rail and sill. The leakage obtained in this run was the leakage entering the window at the frame joints, and at the rubber seal, and that passing through the partition itself. This leakage was subtracted from the total leakage in the locked and not-locked runs to secure the net leakage occurring through the sash perimeter. The leakage in this sealed run is the leakage of the test set-up obtaining when a perfect weatherstrip is applied, hence the name of 100 per cent weatherstrip is applied to this run. In all of the tables and curves of this paper, this sealed run leakage has been subtracted except in the runs shown in Fig. 5 which illustrates methods of sealing. It should be noted that only a part of this sealed run leakage is window frame leakage and that there are no values for frame leakage obtainable from a study of the fourteen frames built for the testing of sash perimeter leakage. Later in this paper, frame leakage is discussed as determined on frames built into wood frame and masonry wall sections.

One Hundred Per Cent Weatherstrip

The purpose of a window is to admit light, to allow the entrance of air when it is wanted, and at other times to prevent all air leakage or infiltration. When the joints between the sash and the frame permit no entrance of air, the sash perimeter is perfect in so far as infiltration is concerned. This condition is secured by the sealing of the sash perimeter on the outside of the window.

To approach this condition in actual building con-

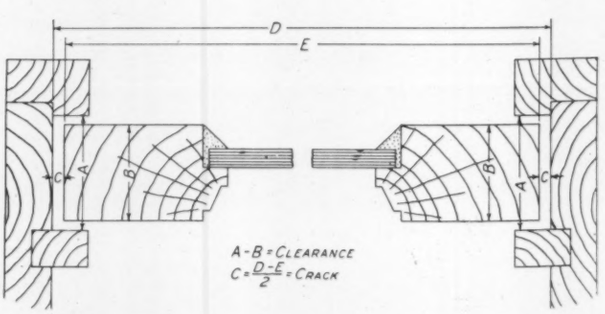

FIG. 3—DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

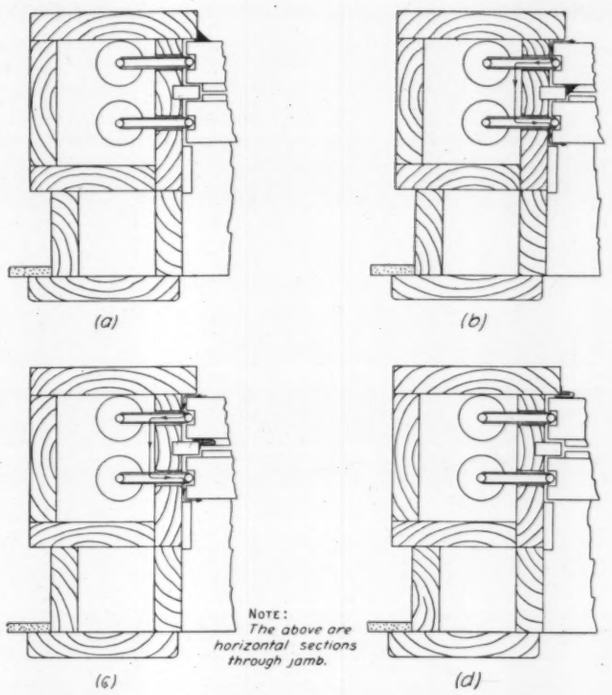

FIG. 4—TWO METHODS OF SEALING SASH PERIMETERS

struction, weatherstripping is generally applied. There are two major factors in weatherstripping that influence the effectiveness of the installation. One of these is the ability of the strip to stop all air that reaches it and the other is the placing of the strip with respect to the sash perimeter joint so that it is in a position to stop all leakage entering this joint. Fig. 4 at *c* and *d* shows the two extreme positions for the placing of the strip. When placed as at *c* on the upper sash, it is possible for air that enters the sash perimeter joint to enter the upper sash pulley holes. When once in the weight spaces, access to the room is gained through the lower sash pulley holes and through cracks in the frame such as at the sash weight doors or around the window trim. Were the strip and its location perfect, it would prevent the entrance of air at the outside contact line between the sash and the frame; the fact that it does not, means that it falls short of being perfect in tightness or in location.

Fig. 4 at *a* and *b* shows the two methods of sealing the sash perimeter to determine the leakage other than sash perimeter leakage. When the seal is applied to the inside of the upper sash as at *b*, it is still possible for air to enter at the sash perimeter and enter the room through the pulley holes and other smaller openings communicating with the sash weight spaces. It is conceivable that an actual weatherstrip applied to the outside of the upper sash as at *d* would stop more leakage than the perfect seal placed on the inner side where it could not prevent air reaching the pulley holes. The actual weatherstrip would then be more perfect than the seal set up as 100 per cent perfect. This obviously indicates a mis-location of the seal and points to the propriety of placing the seal on the outside of the upper sash as well as on the outside of the lower sash.

Fig. 5 shows a comparison of results using the two methods of sealing two plain windows having a high leakage and two weatherstripped windows having a low leakage. The difference between curves *1* and *2* is the leakage obtained for the weatherstripped window perimeter if the seal on the inner side of the upper sash represented 100 per cent weatherstripping. Likewise, the difference between curves *4* and *5* is the leakage through the sash perimeter for the plain window under the same conditions. The differences between curves *2* and *3* and between *5* and *6* represent the leakages that originated in the sash perimeter and found a way into the room through the pulley holes and other smaller openings. This properly should be charged against the sash perimeter whether plain or stripped. It is included as sash perimeter leakage when the seal is made on the outside of both sash. This method of sealing was used in all tests reported in this paper except in the case of two illustrated in Fig. 5 to show the difference in the two methods. In comparing window tests, it is important

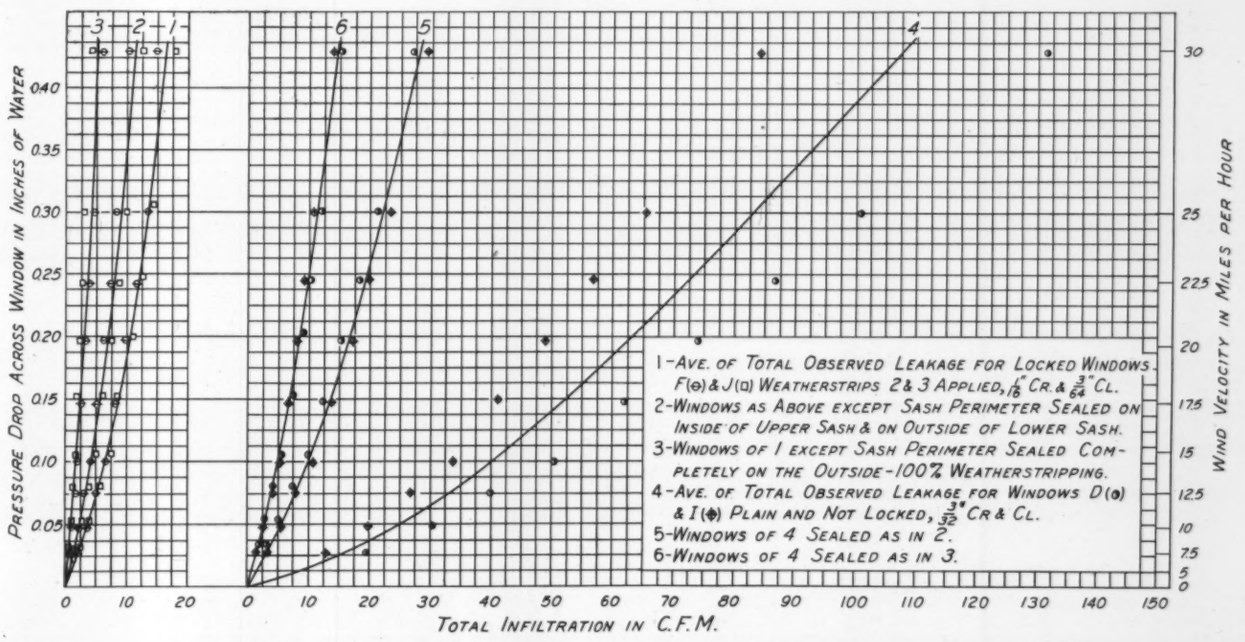
PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
WIND VELOCITY IN MILES PER HOUR
30
25
22.5
20
17.5
15
12.5
10
7.5
5
0
TOTAL INFILTRATION IN C.F.M.
0 10 20 0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150
1-AVE. OF TOTAL OBSERVED LEAKAGE FOR LOCKED WINDOWS. F(○) & J(□) WEATHERSTRIPS 2 & 3 APPLIED, 1/16" CR. & 3/64" CL.
2-WINDOWS AS ABOVE EXCEPT SASH PERIMETER SEALED ON INSIDE OF UPPER SASH & ON OUTSIDE OF LOWER SASH.
3-WINDOWS OF 1 EXCEPT SASH PERIMETER SEALED COMPLETELY ON THE OUTSIDE-100% WEATHERSTRIPPING.
4-AVE. OF TOTAL OBSERVED LEAKAGE FOR WINDOWS D(○) & I(●) PLAIN AND NOT LOCKED, 3/32" CR & CL.
5-WINDOWS OF 4 SEALED AS IN 2.
6-WINDOWS OF 4 SEALED AS IN 3.

FIG. 5—COMPARISON OF RESULTS FOR TWO METHODS OF SEALING SASH PERIMETERS

TABLE 1—DESIGNATION OF WINDOWS AND WEATHERSTRIPS

WINDOW GRADE	GOOD			POOR	GOOD			POOR		GOOD			POOR	
WINDOW	A	B	C	D	E	F	G	H	I	J	K	L	M	N
CURVE SYMBOL	◓	◒	◐	◑	○	⊖	⊠	⊕	⊕	□	⊟	⊠	⊕	⊕
WEATHERSTRIP	← 1 →				← 2 →					← 3 →				

to know which method of sealing was used. Fig. 5 shows that the leakage charged against the weatherstrip is only about one-half of that entering the sash perimeter if the seal is applied to the inside of the upper sash in the case of the two weatherstripped windows represented in this figure.

Setting of Crack and Clearance

The five windows of a poor grade *D, H, I, M,* and *N* fitted loosely when received and were fitted to a uniform 3/32 in. crack and clearance before testing. Fig. 3 illustrates crack and clearance. It is important to note that the crack is one-half of the difference of the horizontal width between runways and the horizontal width of the sash. In the judgment of all who tried them, including a building carpenter boss and a construction inspector, these five windows were considerably looser than the average window in buildings. In the opinion of several weatherstripping mechanics, they were as loose as the usual loosely-fitted window they were called upon to weatherstrip in old buildings, although at times they encountered some that fitted more loosely. All sash would rattle badly except a few with sash members warped to such an extent as to prevent rattling. These five poor windows were tested as plain windows when the crack and clearance were first set at 3/32 in. and again six months later without any alteration to the fit except that which occurred in standing. They were then weatherstripped according to the schedule in Table 1 and again tested.

The nine windows, *A, B, C, E, F, G, J, K,* and *L,* of a good grade were received fitted as closely as possible for free movement of the sash in opening and closing the window. They were tested after a standing period of three months. The crack was found to be approximately 1/64 in. and the clearance 1/32 in. By the definitions of crack and clearance, this means the windows had a tolerance of 1/32 in. in the two directions of sash fit; the difference of the distance between the runways and the width of the sash was 1/32 in. and between the width of the runway and the thickness of the sash was 1/32 in. In the judgment of those trying the windows, they were considered tighter than would be permissible in actual building construction, although under the conditions in the laboratory, they worked easily and smoothly. When they were retested after a standing period of six months from the date of the original tests, it was found that the crack and clearance had increased considerably. The measured crack and clearance were found to average slightly less than 1/16 in. and slightly more than 3/32 in. respectively. They were tested under this condition before a manual change in crack and clearance was made. During the entire 6 months, the windows were sheltered in the laboratory and were carefully handled in the few feet of moving to and from the machine and storage space. The original tests were made in a long dry period of the late summer, and the final tests were made under late winter and early spring heating conditions. After the original tests were completed and before the beginning of the heating season there was a rainy period when the humidity was considerably higher. The increase in crack and clearance during the 6 months' period was observed to be largely an opening up of joints in the frame, which increased the frame leakage and also allowed leakage that started at the outside of the sash perimeter to enter into the weight spaces. Some of this opening of frame joints was considered to be due to the warping of frame members and some to the effect on the frame members of the warping of the closely fitted sash members.

The five loosely fitted windows of a poor grade (*D, H, I, M,* and *N*) showed very little increase in crack and clearance and only a slight increase in leakage during the six months' standing period. This was considered to be due to the sash and frame members having sufficient room to warp without opening up joints and to the longer aging period of one year instead of three months between the time of entering the laboratory to the time of testing. Before the final cracks and clearances were set for the final plain and the weatherstripped tests, the frame joints were closed by using larger nails with heads to supplement the finishing nails originally used.

Since in the judgment of those who tried the windows the 3/32 in. crack and clearance of the five windows of a poor grade were more than the average for old building construction, it was decided to actually measure the fit of a large number of windows. A total of 579 plain double hung wood windows was measured on buildings at least five years old by about a dozen observers within a 300-mile radius of Madison. On an average, 16 were taken on each building, four on each side. The readings were taken in December and January and therefore should represent cracks and clearances existing in the heating season. Eight of the men were assistants in the infiltration testing work and received personal instruction, and every observer received a complete and carefully worded description of the requirements and the method to be followed so that all values would be comparable. A blue-print defining crack and clearance, as does Fig. 3, accompanied the instructions as did mimeographed blanks to be used for the recording of data. Each observer was given a set of thickness gages for determining the clearance. The instructions emphasized the need for watching the warping of sash in measuring clearance and that the lack of clearance at the end of the meeting rails against the parting stops must be compensated for to obtain the full value of the crack. Particular emphasis was placed on the fact that the crack exists on both sides of the sash at the same time as shown in Fig. 3. The average values obtained were 1/16 in. crack and 3/64 in. clearance. The values obtained were slightly less than this for the upper sash and slightly greater for the lower sash. Table 2 shows a summary of the cracks and clearances obtained. The crack and clearance on the series of nine windows were

TABLE 2—FIELD MEASUREMENTS OF CRACK AND CLEARANCE IN INCHES

LOCALITY	NUMBER OF WINDOWS	AVERAGE FOR UPPER SASH		AVERAGE FOR LOWER SASH		MEETING RAIL	TYPE AND NUMBER OF BUILDINGS
		CRACK	CLEARANCE	CRACK	CLEARANCE		
MINNEAPOLIS, MINN.	91	0.0362	0.0465	0.0531	0.0520	0.0757	3 UNIV. - 2 RES.- 1 APT.
HARTLAND, WIS.	42	0.0395	0.0290	0.0603	0.0372	0.0471	5 RESIDENCES.
OCONOMOWOC, WIS.	16	0.0615	0.0415	0.0894	0.0264	0.0283	1 OFFICE
MILWAUKEE, WIS.	32	0.0220	0.0340	0.0720	0.0394	—	2 RESIDENCES
MILWAUKEE, WIS.	80	0.0984	0.0531	0.0765	0.0484	—	1 HIGH SCHOOL GROUP
MADISON, WIS.	16	0.0550	0.0527	0.0840	0.0733	—	1 UNIVERSITY
MADISON, WIS.	96	0.0373	0.0339	0.0659	0.0635	0.0804	6 UNIVERSITY
MADISON, WIS.	64	0.0373	0.0594	0.0694	0.0655	0.0847	1 UNIV.-3 FRATERNITY
WATERFORD, WIS.	46	0.0615	0.0508	0.0544	0.0382	0.0577	3 RESIDENCES
MADISON, WIS.	64	0.0553	0.0510	0.0776	0.0905	0.0992	4 UNIVERSITY
MADISON, WIS.	32	0.0385	0.0437	0.0553	0.0609	0.0515	2 FRATERNITY
WEIGTED AVERAGE	TOTAL 579	0.0500	0.0455	0.0666	0.0567	0.0734	
EQUIVALENT IN FRACTIONS		3/64+	3/64-	1/16+	1/16-	5/64-	
CR& CL. USED ON 9 WINDOWS		1/16	3/64	1/16	3/64	—	

accordingly adjusted to these values of 1/16 in. and 3/64 in. before the final plain and weatherstripped tests were made.

Weatherstripping

After the crack and clearance had been set to the final values and the leakage of the plain windows determined, the windows were weatherstripped by local mechanics representing the weatherstrips they installed. The three mechanics for the three strips were men that had been at such work for many years and had had at least several years' experience with the strip they installed. Two interlocking and one rib type of strip were represented. The mechanics were instructed to install the strips just as they would in actual building construction and it is believed that they did this. The average time to install one strip was 1½ hours and the deviation from this was little. It was not necessary to increase the crack or clearance at the sides of the sash except in a few cases in order to square up the sash.

Presentation of Results

The results of the tests on the fourteen windows are shown in Tables 3 and 4 and in curve form in Figs. 6 to 13, inclusive. Table 1 explains the system of designating the windows and weatherstrips in the tables and on the curves. The tables in addition to giving the leakages at pressures corresponding to wind velocities up to 30 mph also give the pull in pounds necessary to open and close each sash in every test. These pulls give some indication of how tight a plain window has been fitted and the increase in sash pulls upon weatherstripping gives an indication of how closely a strip has been fitted. In some cases, a strip might be made very effective but at an undue loss of ease in opening and closing.

The tables present the values for individual windows and also averages for all of the windows of the two series termed as of good grade and poor grade to which each type of weatherstrip was applied and also the average of the entire groups of five and nine of the two

TABLE 3—RESULTS OF TESTS ON FIVE WINDOWS WITH 3/32 IN. CRACK AND CLEARANCE, PLAIN AND WEATHERSTRIPPED, IN CUBIC FEET PER HOUR, PER FOOT OF SASH PERIMETER

	WINDOW	CRACK	CLEARANCE	SASH PULLS (POUNDS) UPPER UP	UPPER DN.	LOWER UP	LOWER DN.	AVERAGE FOR 1ST AND 2ND CLOSURES — WIND VELOCITY - MILES PER HOUR 5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0	LOCKED — WIND VELOCITY - MILES PER HOUR 5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0
SEPT. TESTS	D	3/32	3/32	5	2	5	0	36.6	65.5	98.7	131.1	164.7	198.6	231.2	262.0	294.3	357.7	13.4	30.0	51.7	76.6	102.8	131.8	161.8	192.3	223.2	271.0
	H	3/32	3/32	2	5	5	0	29.3	52.6	77.8	102.8	127.5	152.2	176.1	199.2	222.7	268.2	19.4	33.7	48.9	64.9	79.4	92.8	104.6	116.0	127.2	148.9
	I	3/32	3/32	6	3	7	2	28.0	48.0	67.7	87.2	106.7	126.6	146.5	165.8	186.1	228.2	18.0	32.9	47.4	62.3	76.8	91.7	106.5	120.8	136.0	166.6
	M	3/32	3/32	5	2	4	2	23.0	43.3	64.6	86.8	109.7	131.5	153.2	175.4	198.6	246.8	11.1	23.1	37.4	53.7	71.2	89.4	108.3	128.0	148.5	190.4
	N	3/32	3/32	6	1	5	1	24.7	42.6	61.8	80.6	99.0	117.6	135.7	154.7	174.8	218.2	10.0	20.9	34.9	50.3	66.5	84.6	102.0	118.9	136.0	170.0
	AVE. H & I	3/32	3/32	4	4	6	1	28.7	50.3	72.8	95.0	117.1	139.4	161.3	182.5	204.4	248.2	18.7	33.3	48.2	63.6	78.1	92.3	105.6	118.4	131.6	157.8
	AVE. M & N	3/32	3/32	5	1	4	1	23.9	43.0	63.2	83.7	104.4	124.5	145.0	165.0	186.7	232.5	10.6	22.0	36.2	52.0	68.9	87.0	105.2	123.5	142.3	180.2
	AVE. D,H,I,M,N	3/32	3/32	5	3	5	1	28.3	50.4	74.0	97.7	121.5	145.3	168.5	191.4	215.3	263.8	14.4	28.1	44.1	61.4	79.3	98.1	116.6	135.2	154.2	189.4
6 MO. LATER	D	3/32	7/64	5	2	6	0	44.0	74.3	107.8	137.2	168.6	203.7	235.8	268.3	302.8	375.2	28.0	52.3	80.3	103.4	129.4	157.5	187.2	219.0	254.3	332.0
	H	3/32	3/32	4	3	6	0	35.6	62.0	88.2	113.9	139.4	164.4	190.4	213.5	244.0	304.4	16.6	33.2	52.6	68.6	86.0	103.7	122.3	141.2	161.2	205.0
	I	3/32	3/32	8	10	10	2	23.3	42.4	62.2	82.0	102.0	122.8	144.3	166.8	189.4	240.5	17.4	32.0	49.2	65.4	81.7	98.8	116.8	135.4	155.7	200.9
	M	3/32	3/32	5	2	4	2	21.4	42.9	64.9	87.1	108.3	129.7	149.7	170.0	191.2	235.5	15.7	32.6	50.8	70.3	89.2	107.4	124.2	140.6	157.1	191.1
	N	3/32	3/32	4	4	5	1	30.3	49.4	68.9	86.9	104.3	121.2	138.0	156.0	174.3	214.7	16.3	33.2	52.6	72.0	91.7	111.2	129.8	148.3	166.9	210.8
	AVE. H & I	3/32	3/32	6	6	8	1	29.5	52.2	75.2	98.0	120.7	143.6	167.4	189.9	216.7	272.5	17.0	32.6	50.9	67.0	83.9	101.3	119.6	138.3	158.5	203.0
	AVE. M & N	3/32	3/32	4	3	4	1	25.9	46.2	66.9	87.0	106.3	125.5	143.9	163.0	182.6	225.2	16.0	32.9	51.7	71.2	90.5	109.3	127.0	144.5	162.0	201.0
	AVE. D,H,I,M,N	3/32	3/32	5	4	6	1	30.9	54.2	78.5	101.4	124.5	148.3	171.8	194.9	220.3	274.1	18.8	36.7	57.1	75.9	95.6	115.7	136.1	156.9	179.0	228.0
WEATHERSTRIPPED	D			14	11	11	6	5.4	11.4	19.2	27.9	37.0	46.7	56.2	66.0	75.7	96.6	5.1	11.1	18.9	27.4	36.6	46.0	55.2	64.8	74.0	93.4
	H			12	9	14	6	5.7	12.7	18.0	25.0	30.6	37.7	45.2	53.6	63.1	83.4	5.7	12.7	18.0	25.0	30.6	37.7	45.2	53.6	63.1	83.4
	I			30	24	14	14	4.0	8.6	14.3	20.9	27.1	34.6	41.7	48.9	56.3	71.4	4.0	8.6	14.3	20.9	27.1	34.6	41.7	48.9	56.3	71.4
	M	AS	ABOVE	24	24	18	15	4.3	8.9	14.0	19.5	24.7	30.4	36.1	42.2	48.3	62.0	4.0	8.3	13.4	18.6	24.0	29.4	34.9	40.9	46.9	60.6
	N			19	20	15	8	4.0	8.6	14.0	20.0	25.9	32.2	38.9	46.1	53.6	69.5	3.7	8.3	13.4	19.2	25.2	31.2	37.4	44.0	51.4	66.0
	AVE. H & I			21	16	14	10	4.9	10.7	16.2	23.0	28.9	36.2	43.5	51.3	59.7	77.4	4.9	10.7	16.2	23.0	28.9	36.2	43.5	51.3	59.7	77.4
	AVE. M & N			22	22	17	11	4.2	8.7	14.0	19.7	25.3	31.3	37.5	44.1	51.0	65.5	3.9	8.3	13.4	18.9	24.6	30.3	36.2	42.5	49.2	63.3
	AVE. D,H,I,M,N			20	17	15	10	4.7	10.0	15.9	22.7	29.1	36.3	43.6	51.4	59.4	76.5	4.5	9.8	15.6	22.2	28.7	35.8	42.9	50.5	58.4	75.0

TABLE 4—RESULTS OF TESTS ON NINE WINDOWS WITH 1/64 IN. CRACK AND 1/32 IN. CLEARANCE AND WITH 1/16 IN. CRACK AND 3/64 IN. CLEARANCE, PLAIN AND WEATHERSTRIPPED, IN CUBIC FEET PER HOUR, PER FOOT OF SASH PERIMETER

	WINDOW	CRACK	CLEARANCE	SASH PULLS POUNDS UPPER UP	UPPER DN	LOWER UP	LOWER DN	AVERAGE FOR 1ST AND 2ND CLOSURES — WIND VELOCITY – MILES PER HOUR 5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0	LOCKED — WIND VELOCITY – MILES PER HOUR 5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0
SEPT. TESTS – ORIG. FIT.	A	1/64	1/32	8	9	7	6	2.8	5.8	9.4	13.4	17.8	22.3	26.9	31.6	36.5	46.2	2.6	5.4	8.9	12.9	17.2	21.6	26.0	30.6	35.3	45.2
	B	1/32–	1/32+	7	8	2	3	2.0	4.6	8.0	11.9	16.0	23.6	25.7	31.1	36.9	49.4	2.2	5.2	8.9	13.2	17.9	23.0	28.5	34.3	40.5	53.5
	C	1/64	1/32+	15	11	7	8	2.1	4.5	7.7	11.3	15.4	19.8	24.4	28.6	33.6	43.3	1.9	4.3	7.4	11.0	15.1	19.5	24.0	28.6	33.2	42.8
	E	1/64+	1/32	5	3	3	4	3.3	6.8	10.8	15.5	20.4	25.8	31.5	37.6	44.0	56.9	3.3	6.8	10.8	15.5	20.4	25.8	31.4	37.3	43.5	56.2
	F	1/64	1/32	7	6	5	2	4.2	8.9	14.4	20.5	27.1	34.3	42.2	50.4	59.2	73.3	4.4	9.2	14.8	21.0	27.7	34.9	42.0	48.8	55.8	67.3
	G	1/64	1/32	16	20	4	6	1.6	3.6	6.2	9.2	12.7	16.6	20.5	24.8	29.3	38.7	1.0	2.6	4.6	7.1	9.7	12.9	16.3	20.0	23.8	32.0
	J	1/64-	1/32	24	24	11	10	2.0	4.5	7.7	11.5	15.5	19.9	24.4	29.1	34.0	44.0	1.7	3.9	6.8	10.3	14.1	18.2	22.5	26.9	31.5	41.1
	K	1/64-	1/32	5	7	8	8	4.2	8.4	13.1	18.3	23.8	29.4	35.1	40.9	47.0	60.1	4.2	8.4	13.1	18.3	23.8	29.4	35.1	40.9	47.0	60.1
	L	1/64	1/32	14	15	36	34	3.1	6.3	10.2	14.7	19.5	24.6	29.8	35.2	40.8	53.1	3.3	6.6	10.6	15.2	20.2	25.3	30.6	36.1	41.8	54.3
	AVE. A,B,C	1/64+	1/32+	10	9	5	6	2.3	5.0	8.4	12.2	16.4	21.9	25.7	30.4	35.7	46.3	2.2	5.0	8.4	12.3	16.7	21.4	26.2	31.2	36.3	47.2
	AVE. E,F,G	1/64+	1/32–	9	10	4	4	3.0	6.4	10.5	15.1	20.1	25.6	31.4	37.6	44.2	56.3	2.9	6.3	10.1	14.5	19.4	24.5	29.9	35.3	41.0	51.8
	AVE. J,K,L	1/64–	1/32	14	15	19	17	3.1	6.4	10.3	14.8	19.6	24.6	29.8	35.1	40.6	52.4	3.1	6.3	10.2	14.6	19.4	24.3	29.4	34.6	40.1	51.8
	AVE. C,G,L	1/64	1/32+	15	15	16	16	2.2	4.8	8.0	11.7	15.9	20.3	24.9	29.5	34.5	45.0	2.1	4.5	7.5	11.1	15.0	19.2	23.6	28.2	32.9	43.0
	AVE. A,B,C,E,F,G,J,K,L	1/64	1/32	11	11	9	9	2.8	5.8	9.6	14.0	18.7	24.0	29.0	34.4	40.2	51.7	2.7	5.9	9.6	13.8	18.5	23.4	28.5	33.7	39.1	50.2
6 MO. LATER	C	1/32	3/64	22	20	12	10	4.6	9.8	15.4	21.4	27.4	34.0	42.0	49.7	58.0	76.3	3.4	7.6	12.9	18.0	24.6	30.6	38.6	46.3	54.0	72.3
	G	1/16	1/16	4	4	4	6	3.7	7.9	13.8	21.0	28.9	36.2	45.7	55.2	65.0	85.9	2.3	6.0	10.9	17.2	23.7	30.3	38.9	47.5	56.6	75.7
	L	5/64	3/64	7	7	16	16	5.0	11.3	18.5	26.0	33.0	40.9	49.5	57.8	66.6	85.6	5.1	11.7	18.9	26.6	33.7	41.4	50.0	58.3	67.2	86.6
	AVE. C,G,L	1/16–	3/64+	11	10	11	11	4.4	9.6	15.9	22.8	29.8	37.0	45.7	54.2	63.2	82.6	3.6	8.4	14.2	20.6	27.3	34.1	42.5	50.7	59.3	78.2
FINAL FIT	A	1/16	3/64	5	6	2	4	6.9	14.6	23.4	33.2	43.2	53.7	63.7	73.7	84.0	104.3	8.0	16.3	25.1	35.4	45.4	56.0	66.3	76.6	86.6	106.8
	B	1/16	3/64	4	5	2	4	5.0	10.3	16.8	23.5	31.1	38.2	47.6	56.3	65.6	87.2	5.0	10.6	17.2	23.7	31.5	38.9	48.3	57.2	66.6	87.4
	C	1/16	3/64	16	16	8	10	6.6	13.2	20.5	27.9	35.6	43.6	52.1	60.5	69.3	88.2	6.3	12.3	19.3	26.4	34.0	41.9	50.2	58.3	67.1	85.7
	E	1/16	3/64	4	2	3	3	5.1	11.9	19.6	28.5	38.0	47.3	57.2	67.1	77.3	99.6	5.6	12.3	20.3	30.0	40.6	51.5	62.3	73.1	84.9	108.9
	F	1/16	3/64	6	4	6	5	7.4	14.9	24.0	33.7	44.0	53.4	65.2	75.7	86.6	108.3	8.6	16.9	26.6	36.6	46.9	56.3	68.0	78.3	88.6	109.1
	G	1/16	3/64	4	5	2	3	3.7	7.7	12.7	18.5	24.7	31.4	38.4	45.4	53.0	69.2	3.2	7.4	11.8	17.0	22.7	28.8	35.2	42.0	49.2	64.6
	J	1/16	3/64	10	12	8	10	6.2	12.5	20.5	29.1	38.8	48.8	58.5	69.0	79.6	101.6	4.0	8.3	14.7	21.2	29.2	37.7	46.3	55.4	65.2	86.0
	K	1/16	3/64	4	5	3	6	4.6	9.2	16.3	24.0	32.1	39.5	48.9	57.7	66.3	84.5	4.3	8.9	16.0	23.4	31.5	38.6	47.7	56.0	64.0	81.2
	L	1/16	3/64	8	8	28	28	4.6	11.0	18.3	25.6	33.2	40.9	48.6	56.9	65.4	84.0	4.6	11.0	18.3	25.6	33.2	40.9	48.6	56.9	65.4	84.0
	AVE. A,B,C	1/16	3/64	8	9	4	6	6.2	12.7	20.2	28.2	36.6	45.2	54.5	63.5	73.0	92.9	6.4	13.1	20.5	28.5	37.0	45.6	54.9	64.0	73.4	93.2
	" E,F,G	1/16	3/64	5	4	4	4	5.4	11.5	18.8	26.9	35.6	44.0	53.6	62.7	72.3	92.4	5.8	12.2	19.6	27.9	36.7	45.5	55.2	64.5	74.2	94.2
	" J,K,L	1/16	3/64	7	8	13	15	5.1	10.9	18.4	26.2	34.6	43.1	52.0	61.2	70.4	90.0	4.3	9.4	16.3	23.4	31.3	39.1	47.5	56.1	64.9	83.7
	AVE. A,B,C,E,F,G,J,K,L	1/16	3/64	7	7	7	8	5.6	11.7	19.1	27.1	35.6	44.1	53.4	62.5	71.9	91.8	5.5	11.8	18.8	26.6	35.0	43.4	52.5	61.5	70.8	90.4
WEATHERSTRIPPED	A	1/16	3/64	33	32	10	10	2.2	3.9	6.4	9.1	12.0	14.9	18.0	21.3	25.0	32.7	1.6	3.4	5.5	7.9	10.2	12.7	15.5	18.3	21.4	28.1
	B	1/16	3/64	16	18	16	15	1.8	3.8	6.4	9.3	12.4	15.7	19.1	22.7	26.5	34.5	1.5	3.2	5.5	8.0	10.6	13.6	16.5	19.5	22.7	29.4
	C	1/16	3/64	22	22	13	12	2.1	4.4	7.2	10.3	13.8	17.2	20.7	24.1	27.8	35.5	1.7	3.5	5.7	8.4	11.3	14.2	17.2	20.2	23.5	30.4
	E	1/16	3/64	14	14	9	7	3.4	6.9	10.5	14.9	19.2	23.6	28.2	32.5	37.4	47.9	3.1	6.3	9.9	14.0	18.3	22.7	27.1	31.3	35.9	45.4
	F	1/16	3/64	18	16	12	9	4.1	8.3	13.0	17.7	22.5	27.4	32.6	37.9	43.8	56.8	3.6	7.6	12.1	17.2	21.7	26.0	30.7	35.4	40.6	51.6
	G	1/16	3/64	16	16	9	12	3.0	6.3	10.1	14.0	18.3	22.3	26.6	31.2	35.5	45.6	3.6	7.1	11.1	15.2	19.7	24.0	28.3	32.9	37.2	47.7
	J	1/16	3/64	22	24	16	15	2.8	5.7	8.7	12.7	16.6	19.9	24.2	28.3	32.4	40.7	2.6	5.4	8.3	12.3	16.0	19.2	23.4	27.4	31.4	39.7
	K	1/16	3/64	18	20	11	10	2.7	5.3	8.5	11.7	15.1	18.0	21.8	25.5	29.4	38.4	2.4	4.7	8.2	11.5	14.7	17.8	21.6	25.4	29.5	38.9
	L	1/16	3/64	17	15	22	20	1.9	4.0	6.7	10.3	13.9	17.9	22.0	26.0	30.5	35.2	1.7	3.4	6.0	9.1	12.6	16.3	20.0	23.7	28.3	37.4
	AVE. A,B,C	1/16	3/64	24	24	13	13	2.0	4.0	6.6	9.6	12.7	15.9	19.3	22.7	26.4	34.2	1.6	3.4	5.6	8.1	10.7	13.5	16.4	19.3	22.5	29.3
	" E,F,G	1/16	3/64	18	15	10	9	3.5	7.2	11.2	15.5	20.0	24.4	29.1	33.9	38.9	50.1	3.4	7.0	11.0	15.5	19.9	24.2	28.7	33.2	37.9	48.2
	" J,K,L	1/16	3/64	19	20	16	15	2.5	5.0	8.0	11.6	15.2	18.6	22.7	26.6	30.8	39.8	2.2	4.5	7.5	11.0	14.4	17.8	21.7	25.5	29.7	38.7
	AVE. A,B,C,E,F,G,J,K,L	1/16	3/64	20	20	13	12	2.7	5.4	8.6	12.2	16.0	19.6	23.7	27.7	32.0	41.4	2.4	5.0	8.1	11.5	15.0	18.5	22.3	26.0	30.1	38.7

series. Although two not-locked runs, termed 1st and 2nd closures, were made in each case, the tables show only the average of the two. The curve sheets that show the points for individual windows and the averages of groups as curves show points for both the 1st and 2nd closures except where they were identical. However, in getting values for average curves all test values, whether identical or not, were given equal weight. This explanation is considered necessary because some of the curves do not appear to be the average of the points they represent because some runs are represented by one point and others by two points at each pressure reading.

Table 3 shows the results of the tests on five windows of a poor grade with 3/32 in. crack and clearance, both plain and stripped. Table 4 shows the results for the nine windows termed as of a good grade both plain with the original 1/64 in. crack and 1/32 in. clearance and plain and weatherstripped with the final 1/16 in. crack and 3/64 in. clearance that were established as representing the average crack and clearance by field measurements on a large number of windows.

Fig. 6 presents the results for the five windows plain and weatherstripped with 3/32 in. crack and clearance, not locked. Fig. 7 is the same for the locked runs. Figs. 8 and 9 show the corresponding results for the nine windows with 1/16 in. crack and 3/64 in. clearance. These four figures show the points by individual windows and the average curves by weatherstrips.

Fig. 10 shows the variation in infiltration over the standing period of 6 months between the original and the check tests without the making of a manual alteration in the crack and clearance. Fig. 11 shows the average curves for the series of five windows using 3/32 in. crack and clearance under locked and not-locked, plain and weatherstripped conditions. Fig. 12 shows the average curves for the series of nine windows with the original 1/64 in. crack and 1/32 in. clearance and with the final 1/16 in. crack and 3/64 in. clearance, both plain and weatherstripped.

Discussion of Results

The average leakage found for the five plain, not-locked windows with 3/32 in. crack and clearance, was 124.5 cfh per foot of sash perimeter at 15 mph. This is the result of the tests made directly before weatherstripping and compares with 29.1 cfh for the same windows weatherstripped with the same crack and clearance. The points on Fig. 6 show that there was considerable variation in leakage between the five plain windows. This is considered to be due largely to the poor workmanship and material of this group. It was difficult to set an even crack and clearance on these five windows.

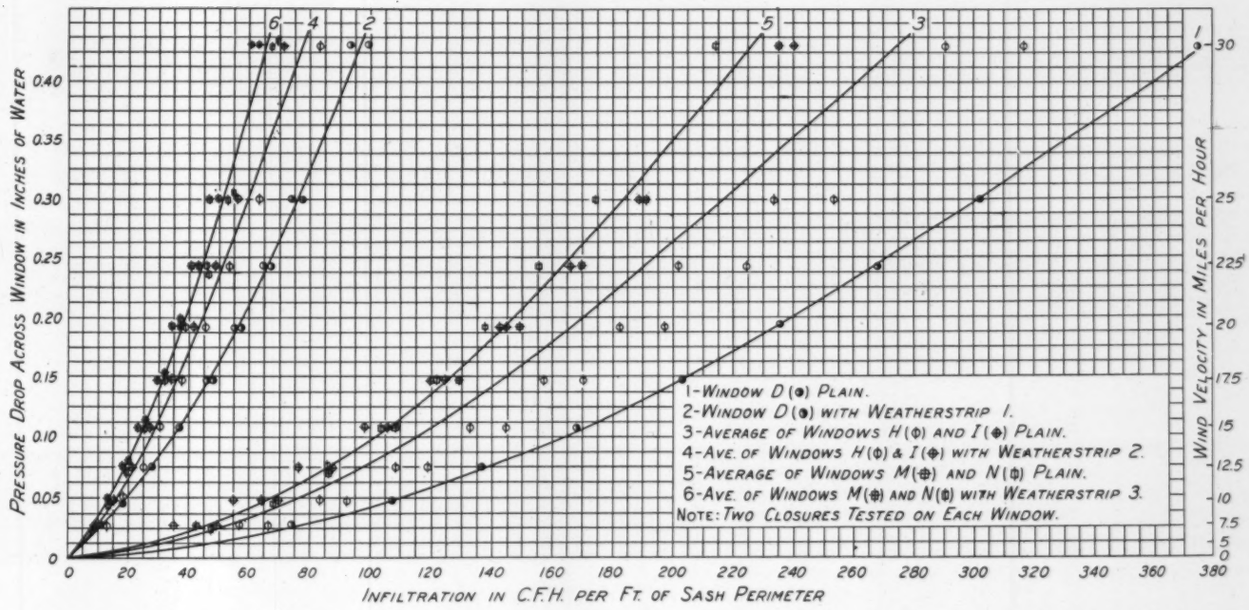
PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER
WIND VELOCITY IN MILES PER HOUR
INFILTRATION IN C.F.H. PER FT. OF SASH PERIMETER
1-WINDOW D (●) PLAIN.
2-WINDOW D (●) WITH WEATHERSTRIP 1.
3-AVERAGE OF WINDOWS H (⊕) AND I (⊕) PLAIN.
4-AVE. OF WINDOWS H (⊕) & I (⊕) WITH WEATHERSTRIP 2.
5-AVERAGE OF WINDOWS M (⊕) AND N (⊕) PLAIN.
6-AVE. OF WINDOWS M (⊕) AND N (⊕) WITH WEATHERSTRIP 3.
NOTE: TWO CLOSURES TESTED ON EACH WINDOW.

FIG. 6—RESULTS FOR FIVE PLAIN AND WEATHERSTRIPPED WINDOWS WITH 3/32 IN. CRACK AND CLEARANCE—NOT LOCKED

Window *D* showed a very high leakage as a plain window due to a combination of an upper sash that was warped and not square so that the head and meeting rail leakages were high and a larger than average clearance. Plain window *H* was also higher in leakage than the average when not locked. Locking reduced the leakage through this window so that windows *H*, *I*, *M*, and *N* had very nearly the same air leakage when locked as shown in Fig. 7. Locking window *D* reduced the leakage somewhat but left its position relative to the average about the same. Although window *D* deviates from the average considerably, it appeared to be an entirely possible window to be in any group of five windows of this type; consequently, it was used in the averaging of the group. The average leakage at 15 mph for this window when plain, for both locked and not locked runs, was 35 per cent above the average of the group. It was 28 per cent above the average of the group when weatherstripped. Windows *H* and *I* were 8 per cent lower than the average of the group as plain windows and were equal to the average as weatherstripped windows. Windows *M* and *N* were 10 per cent lower than the average when plain and were 14 per cent lower when weatherstripped, considering both locked and not-locked results. This seems to indicate that the leakage through the weatherstripped window for this class and fit of window depends largely on the leakage of the original plain window. The curves in Figs. 6 and 7 indicate this relation between the leakage of the plain and weatherstripped windows. Weatherstrip (1) applied to window *D* did not prove relatively as effective as when applied to windows *A*, *B*, and *C* as shown in Fig. 8. It would seem that

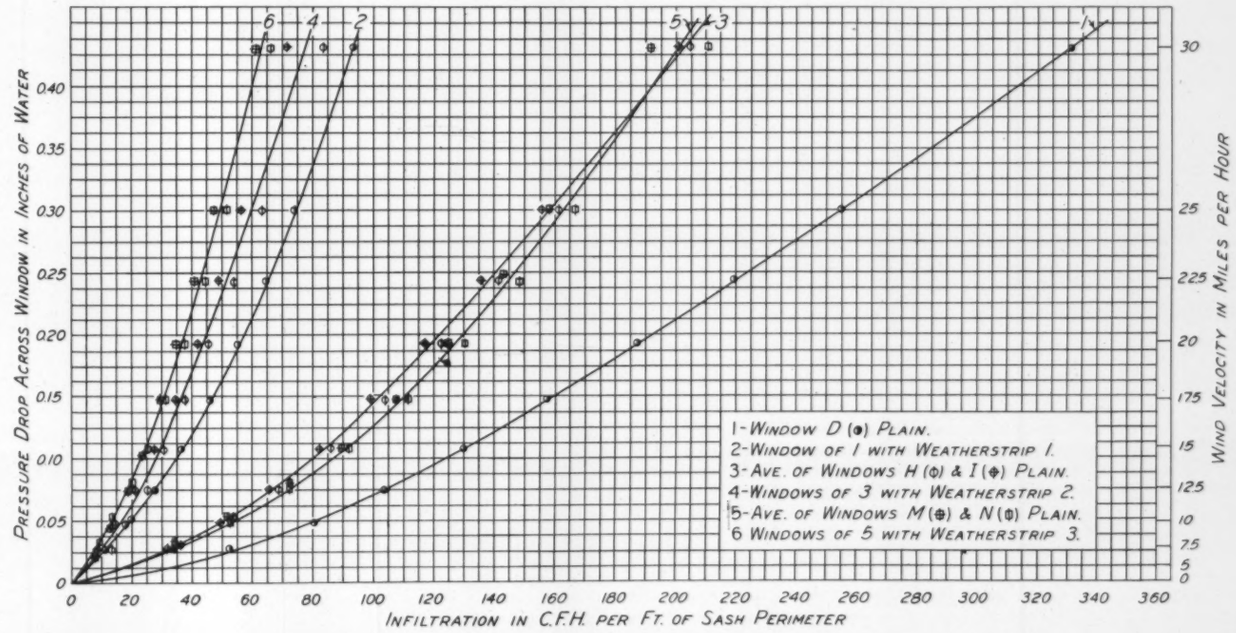
PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER
WIND VELOCITY IN MILES PER HOUR
INFILTRATION IN C.F.H. PER FT. OF SASH PERIMETER
1-WINDOW D (●) PLAIN.
2-WINDOW OF 1 WITH WEATHERSTRIP 1.
3-AVE. OF WINDOWS H (⊕) & I (⊕) PLAIN.
4-WINDOWS OF 3 WITH WEATHERSTRIP 2.
5-AVE. OF WINDOWS M (⊕) & N (⊕) PLAIN.
6 WINDOWS OF 5 WITH WEATHERSTRIP 3.

FIG. 7—RESULTS FOR FIVE PLAIN AND WEATHERSTRIPPED WINDOWS WITH 3/32 IN. CRACK AND CLEARANCE—LOCKED

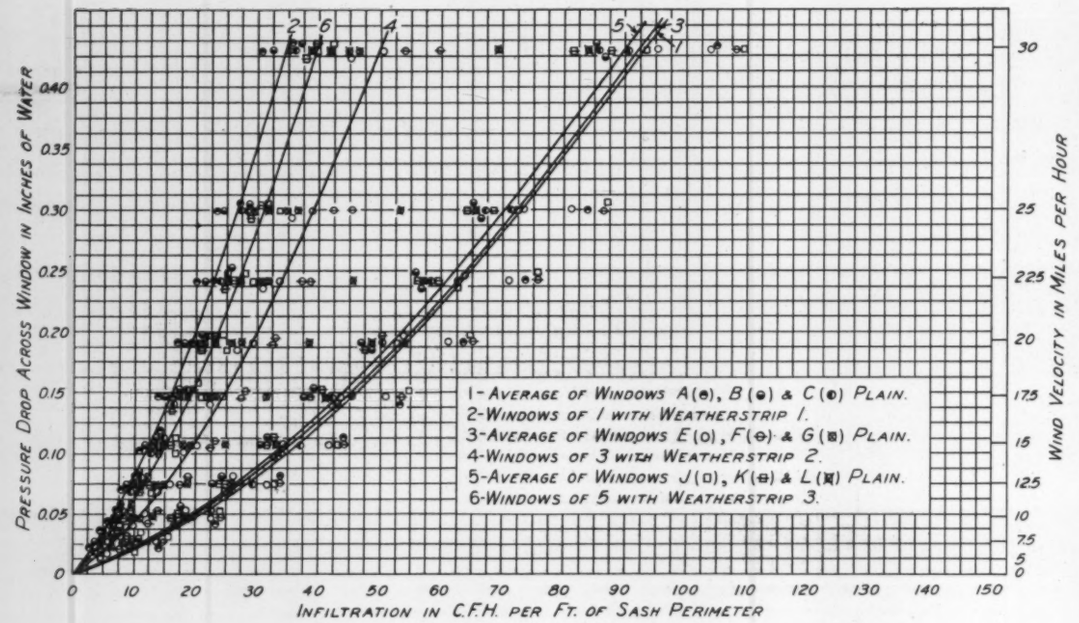
Pressure Drop Across Window in Inches of Water
Wind Velocity in Miles per Hour
Infiltration in C.F.H. per Ft. of Sash Perimeter
1-Average of Windows A(●), B(●) & C(●) Plain.
2-Windows of 1 with Weatherstrip 1.
3-Average of Windows E(○), F(⊖) & G(⊠) Plain.
4-Windows of 3 with Weatherstrip 2.
5-Average of Windows J(□), K(⊟) & L(▣) Plain.
6-Windows of 5 with Weatherstrip 3.

Fig. 8—Results for Nine Plain and Weatherstripped Windows with 1/16 in. Crack and 3/64 in. Clearance—Not Locked

the high leakage of the poorest of the five windows was caused by factors that the weatherstrip could not overcome to the same extent as on the better windows *A*, *B*, and *C*.

An examination of individual points shows that there is a considerable variation between windows *H* and *I*, plain and not locked, to which weatherstrip *2* was applied. Locking reduced the leakage through window *H* to a value very close to that of *I* when locked. Applying weatherstrips to *H* and *I* brought the leakage of these two windows very close together. The points for each weatherstripped window lie very close to the average for the windows to which that strip was applied.

The curves for the three groups of windows, *D* and *H I*, and *M N* in Figs. 6 and 7 show that much more variation exists between the averages for the plain windows than for the weatherstripped windows. At 15 mph the variation is 62 cu ft for the plain windows and 12 cu ft for the weatherstripped windows.

Fig. 11 shows a considerable reduction in the results for the locking of the plain windows. When weatherstripped, locking resulted in very little reduction on these five windows.

The average leakage for the nine plain, not locked windows with 1/16 in. crack and 3/64 in. clearance is shown in curve *3* of Fig. 12 and was 35.6 cu ft per hour per foot of sash perimeter at 15 mph. The greatest variation from this for the groups of three windows

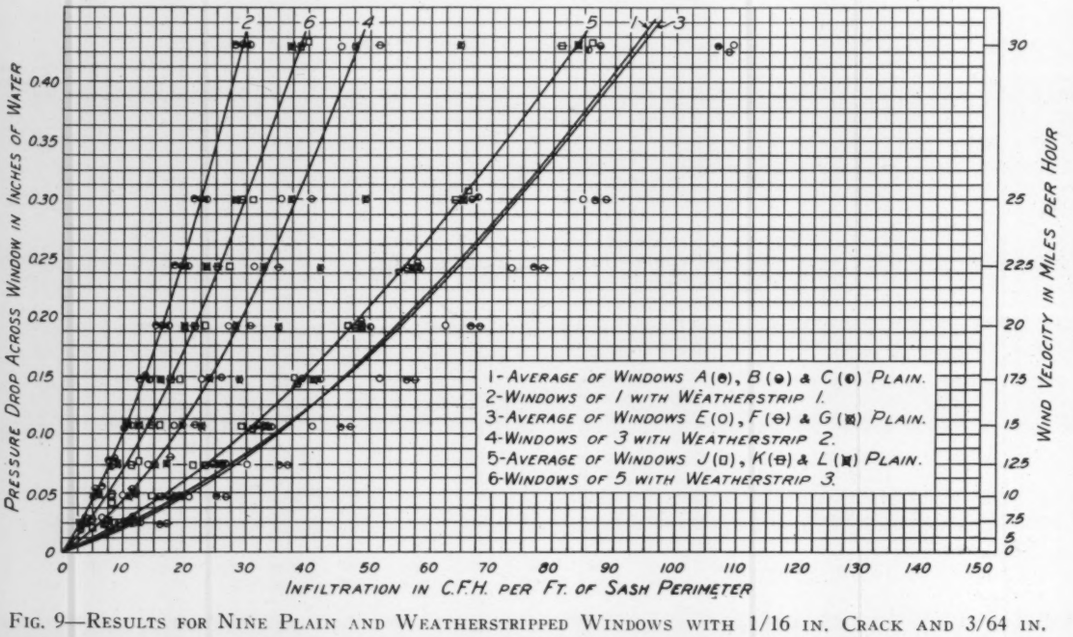
Pressure Drop Across Window in Inches of Water
Wind Velocity in Miles per Hour
Infiltration in C.F.H. per Ft. of Sash Perimeter
1-Average of Windows A(●), B(●) & C(●) Plain.
2-Windows of 1 with Weatherstrip 1.
3-Average of Windows E(○), F(⊖) & G(⊠) Plain.
4-Windows of 3 with Weatherstrip 2.
5-Average of Windows J(□), K(⊟) & L(▣) Plain.
6-Windows of 5 with Weatherstrip 3.

Fig. 9—Results for Nine Plain and Weatherstripped Windows with 1/16 in. Crack and 3/64 in. Clearance—Locked

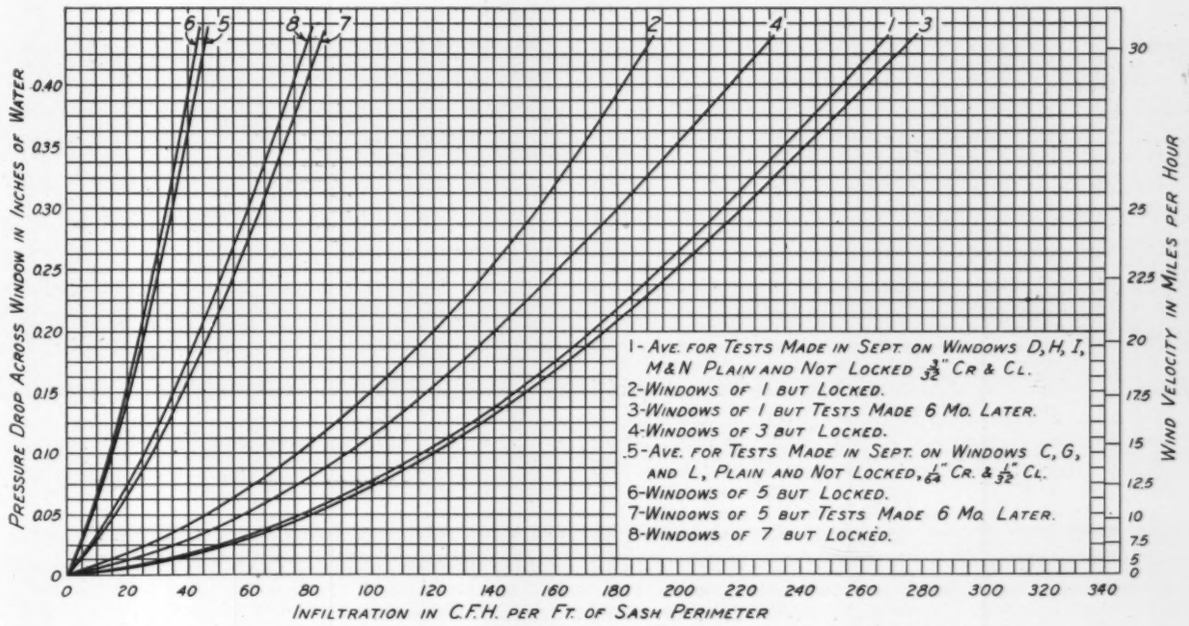
Pressure Drop Across Window in Inches of Water
Wind Velocity in Miles per Hour
Infiltration in C.F.H. per Ft. of Sash Perimeter
1-Ave. for Tests Made in Sept. on Windows D, H, I, M & N Plain and Not Locked 3/32" Cr & Cl.
2-Windows of 1 but Locked.
3-Windows of 1 but Tests Made 6 Mo. Later.
4-Windows of 3 but Locked.
5-Ave. for Tests Made in Sept. on Windows C, G, and L, Plain and Not Locked, 1/64" Cr. & 1/32" Cl.
6-Windows of 5 but Locked.
7-Windows of 5 but Tests Made 6 Mo. Later.
8-Windows of 7 but Locked.

Fig. 10—Variation in Infiltration Over a Standing Period of Six Months, Crack and Clearance Not Altered Manually

was only 3 per cent. The variation for individual windows in the group as shown on Fig. 8 was considerably more. Window *G* had an especially low leakage as a plain window. When weatherstripped, the leakage of this window was one of the highest. Window *A*, plain, was the highest in leakage of the entire nine, but when weatherstripped was the lowest. The same strip (1) was applied to a window (*C*) which as a plain window had about the average leakage of the entire nine and to one with next to the lowest leakage (*B*). When weatherstripped, these three windows with widely varying leakages as plain windows were very uniform in leakage. The same variation is noticed with windows *E*, *F* and *G* to which weatherstrip (2) was applied. As plain windows, the results for the three windows of the group varied widely, yet when weatherstripped, the results group themselves quite closely to the average curve (*4* of Fig. 8). The variations in the three groups as plain windows are about the same except that *G* had considerably lower leakage than *B*, the lowest in the other two groups. When weatherstripped, group *E*, *F*, *G* was highest in leakage and group *A*, *B*, *C* the lowest in leakage. The results of the tests of these weatherstripped windows were closely grouped about their own average group curve (*2*, *6*, and *4*). This close grouping of the weatherstrips about the average curve for each group regardless of the variation of the individual windows in the group as plain windows, seems to indicate that each strip as applied to a window of this average crack and clearance has certain characteristics that determine the

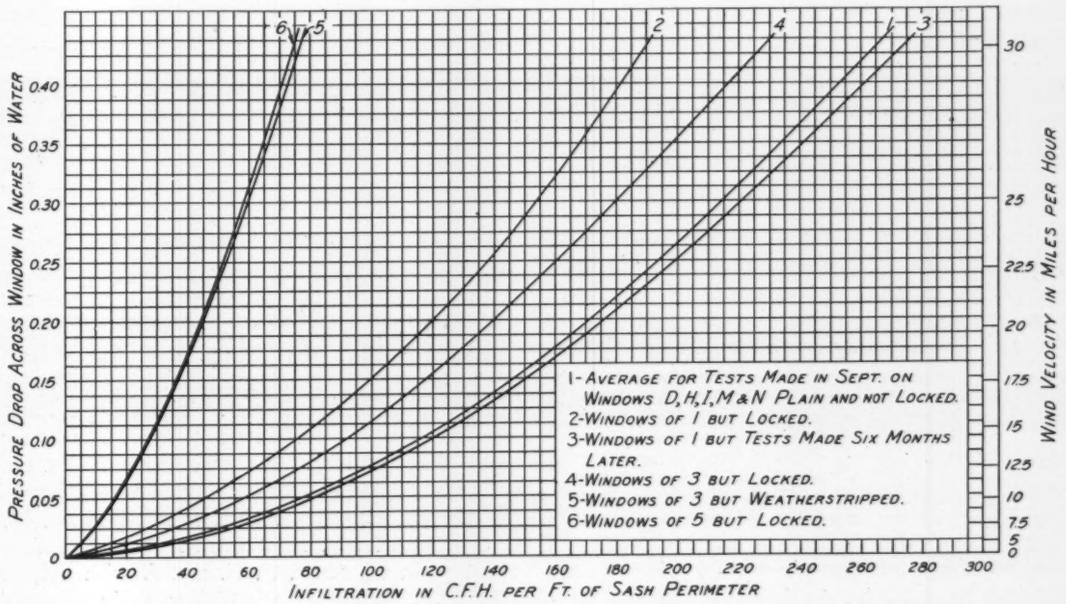
Pressure Drop Across Window in Inches of Water
Wind Velocity in Miles per Hour
Infiltration in C.F.H. per Ft. of Sash Perimeter
1-Average for Tests Made in Sept. on Windows D, H, I, M & N Plain and not Locked.
2-Windows of 1 but Locked.
3-Windows of 1 but Tests Made Six Months Later.
4-Windows of 3 but Locked.
5-Windows of 3 but Weatherstripped.
6-Windows of 5 but Locked.

Fig. 11—Summary Curves for Five Windows with 3/32 in. Crack and Clearance

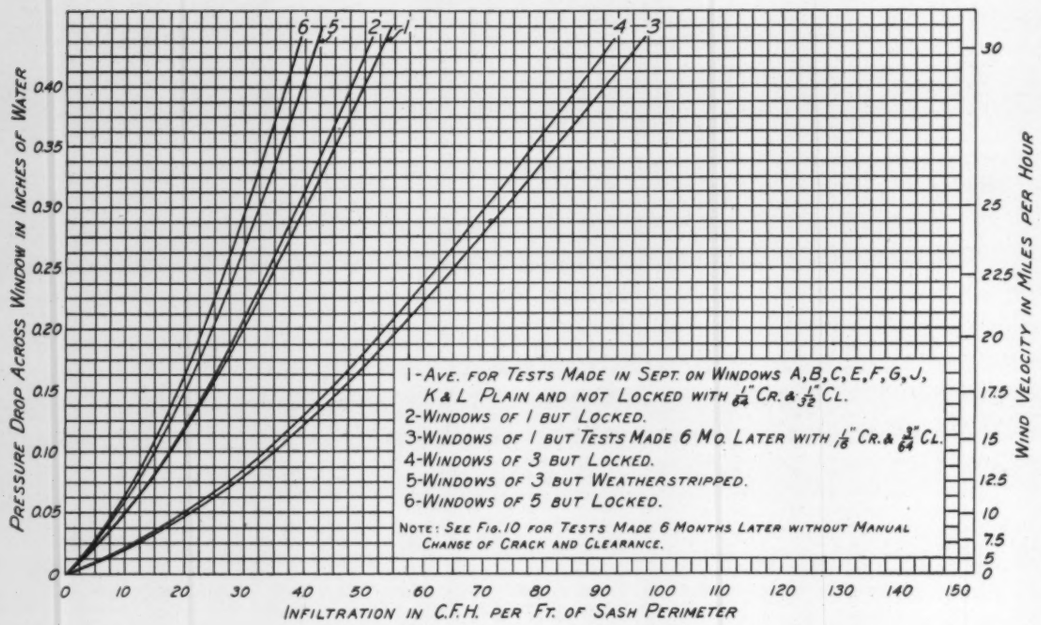
Pressure Drop Across Window in Inches of Water
Wind Velocity in Miles per Hour
Infiltration in C.F.H. per Ft. of Sash Perimeter
1-Ave. for Tests Made in Sept. on Windows A, B, C, E, F, G, J, K & L Plain and not Locked with 1/64" Cr. & 1/32" Cl.
2-Windows of 1 but Locked.
3-Windows of 1 but Tests Made 6 Mo. Later with 1/16" Cr. & 3/64" Cl.
4-Windows of 3 but Locked.
5-Windows of 3 but Weatherstripped.
6-Windows of 5 but Locked.
Note: See Fig. 10 for Tests Made 6 Months Later without Manual Change of Crack and Clearance.

Fig. 12—Summary Curves for Nine Windows with 1/64 in. Crack and 1/32 in. Clearance and with 1/16 in. Crack and 3/64 in. Clearance

weatherstripped window leakage. Had considerable variation been found among the weatherstripped windows of each group, it would have been due likely to a non-uniformity of the strip, to the influence of the plain window fit and leakage, or to the variation in workmanship resulting when the same mechanic applied the same type of strip to several windows.

The grouping of the points for the individual weatherstripped windows *D, H, I, M,* and *N* with 3/32 in. crack and clearance as shown on Fig. 6 is very close to the average curves for *D* and *H I* and *M N,* although the plain windows *H* and *I* varied widely. One difference noted between the application of strips to the two groups of five and nine windows was the relative positions occupied by the plain and stripped results of the windows *A, B, C* and *D* to which strip (1) was applied. *A, B,* and *C* as a group had the highest plain leakage of the nine and as weatherstripped the lowest leakage. Window *D,* plain, had considerably the highest leakage of the five in the series, and when weatherstripped, it still had the highest leakage. It, however, was about 8 per cent nearer the average of the group when weatherstripped than when plain.

The average leakage for the nine weatherstripped windows not locked as shown in Fig. 12, curve 5 was 16.0 cu ft per hour per foot of sash perimeter at 15 mph. The deviation of the average of each of the three groups from the general average of the nine windows when weatherstripped is considerably more than the maximum deviation of 3 per cent when plain. Group *A, B, C* was about 20 per cent lower than the average, group *E, F, G* was 25 per cent higher, and group *J, K, L* 5 per cent lower than the average. This indicates that there is more variation between the three groups when weatherstripped than when plain for these windows fitted to a 1/16 in. crack and 3/64 in. clearance.

The average leakage of the nine plain windows as originally fitted with 1/64 in. crack and 1/32 in. clearance was 18.7 cu ft as compared to 16.0 cu ft for the weatherstripped window with a crack of 1/16 in. and a clearance of 3/64 in. This is shown in Fig. 12. A plain window then can be made with a small crack and clearance and yet will open and close easily, and will have a leakage almost as low as a weatherstripped window fitted with an average crack and clearance. It is not known what the comparison would be if the weatherstrips had been applied to the windows retaining the small crack and clearance. It also must be borne in mind that the plain windows were fitted too closely to permit working under all conditions encountered in actual building constructions.

Locking the nine plain and weatherstripped windows resulted only in a one cubic foot reduction in leakage. The reduction due to locking was also slight in the case of the five windows when weatherstripped. The only case where locking resulted in a substantial reduction in leakage was that of these five plain windows which were fitted to a 3/32 in. crack and clearance. All locked runs were made with the locks just as they were originally fitted at the mill. Locking comparisons are shown on Figs. 11 and 12 for the two series of windows.

Fig. 10 shows the variation in results between the original tests made in September and 6 months later without manually changing the crack and clearance. On the five windows of a poor grade, the crack and clearance remained practically constant. The 6 months later tests on the plain, not-locked windows showed an increase in leakage of less than three per cent. Comparing the locked runs, the leakage of the same windows increased about 20 per cent. The reason for this larger increase for the locked as compared to the not-locked runs is not known. It would seem that a shrinking or warping of the window members would have less effect on the leakage when locked than when not locked.

Curve 5 of Fig. 10 shows the average leakage through three plain windows, *C, G,* and *L* with a 1/64 in. crack and 1/32 in. clearance. After a six months' standing period, the windows had become much looser. The

average crack became slightly less than 1/16 in. and the average clearance slightly over 1/32 in. The average leakage for the plain, not locked windows in the original tests was 15.9 cu ft per hour per foot of sash perimeter at 15 mph. The leakage after the standing period of 6 months was 29.8 cu ft. This is an increase in leakage of 87 per cent. A large part of the increase in clearance was due to the warping and shrinking of the wood pulling loose the finishing nails with which the frames were held together. The driving in of larger nails restored the clearances to somewhere near their original values. The crack and clearance were then adjusted to the final values of 1/16 in. crack and 3/64 in. clearance by planing. The leakage then obtained for the plain, not-locked window was 31.2 cu ft per hour per foot of sash perimeter at 15 mph. Locking resulted in a very small reduction in leakage in these three sets of tests on windows *C*, *G*, and *L*.

Locking resulted in no change in leakage in 20 per cent of all the runs made in these tests of the fourteen windows. In another 20 per cent of the runs, locking increased the leakage and in 60 per cent of the runs, it decreased the leakage. The greatest decrease in leakage was 38 per cent and the greatest increase was 13 per cent. The average of all changes was an 8 per cent reduction.

Comparing the results of the first and second closures, it was found that in 31 per cent of the runs the leakage was the same for both closures. The average variation between the two closures was 5 per cent. The greatest variation was 21 per cent; the variation was greater than 10 per cent in 12 per cent of the runs.

Fig. 13 shows the influence on the infiltration of the location of the weatherstrip on the upper sash. Fig. 4 at *c* shows the strip located on the inside of the upper sash and *d* shows the strip located on the outside of the upper sash. When located as at *c*, it is possible for air entering the sash perimeter to gain entrance to the room through the pulley holes. The only reduction weatherstripping in this position could make in this leakage through the pulley holes would be in decreasing the clearance at the outside joint between the sash and the runway by exerting a force in that direction. The strip represented in Fig. 13 is regularly applied to the inside of the upper sash. The results of the tests of two windows with large crack and clearance weatherstripped in this manner are shown by curve *2* and as applied to three windows of average crack and clearance are shown in curve *5*. After the completion of these tests, the strips were removed from the upper sash and similar strips installed on the outside of this sash. The change in location resulted in a reduced leakage in every case. Curves *3* and *6* show the results for this relocation of the strips. The reduction was 26 per cent for the two windows with large crack and clearance and 37 per cent for the three windows with small crack and clearance. This reduction is due to the reduction of sash perimeter leakage that enters the room through the pulley holes.

Frame Leakage

The results of the tests on the fourteen windows as given in Tables 3 and 4 and Figs. 6 to 13 inclusive are for sash perimeter leakage only. To this leakage must be added any air leakage that occurs through the frame of the window. In a plain plastered wall whether of masonry or frame construction, the leakage on a plain wall area the size of a window opening is negligible. Any leakage over this negligible amount when a window opening is placed in the wall is rightfully chargeable to the window. This leakage in a wood frame wall is mainly due to air entering at the edges of the building paper at the frame opening. A certain amount also enters at the corners of buildings and after getting under the paper travels in the joints of the sheathing from one studding space to another until a window opening, door opening or baseboard is reached that allows entrance to the room.

In a masonry wall, the leakage originates mainly at the joint between the window frame and the masonry wall, known as the frame joint. This leakage can be practically eliminated by the proper use of calking compound. A smaller amount of the frame leakage originates in the

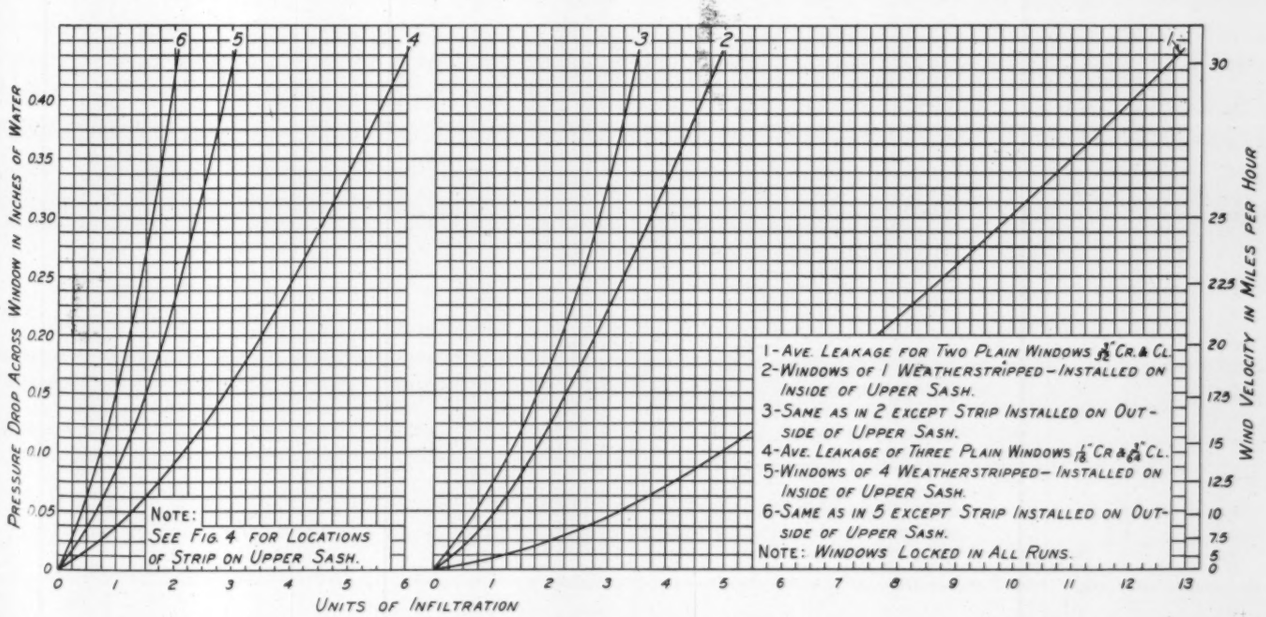
Pressure Drop Across Window in Inches of Water
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
Wind Velocity in Miles per Hour
30
25
22.5
20
17.5
15
12.5
10
7.5
5
0
Units of Infiltration
0 1 2 3 4 5 6 0 1 2 3 4 5 6 7 8 9 10 11 12 13
1-Ave. Leakage for Two Plain Windows 1/32" Cr. & Cl.
2-Windows of 1 Weatherstripped-Installed on Inside of Upper Sash.
3-Same as in 2 except Strip Installed on Outside of Upper Sash.
4-Ave. Leakage of Three Plain Windows 1/16" Cr & 3/64" Cl.
5-Windows of 4 Weatherstripped-Installed on Inside of Upper Sash.
6-Same as in 5 except Strip Installed on Outside of Upper Sash.
Note: Windows Locked in All Runs.
Note:
See Fig. 4 for Locations of Strip on Upper Sash.

Fig. 13—Effect on Infiltration of Strip Location

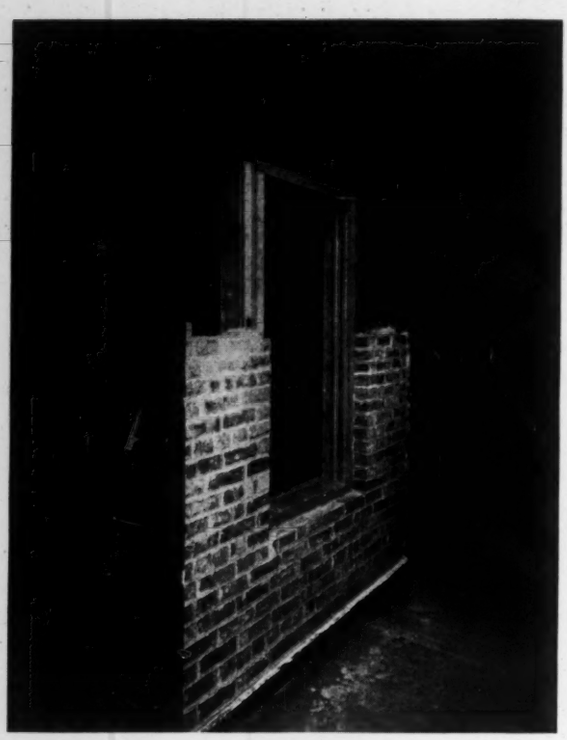

FIG. 14—BOX FRAME IN MASONRY WALL TESTED FOR FRAME LEAKAGE

wall itself and travels horizontally in the voids to the window frame. Calking, of course, does not reduce this leakage. The use of solid masonry, that is, completely slushed joints, rather than the use of incompletely slushed joints that leave voids in the interior of the wall, would reduce this leakage. A portion of the frame leakage enters the room around the interior window trim, and some enters the sash weight boxes from where for the most part it enters the room through the lower sash pulley holes. The total pulley hole leakage is composed of this part of the frame leakage and the part of the sash perimeter leakage that entered the weight boxes mainly through the upper sash pulley holes. Pulley hole covers are at times applied to reduce this leakage. They reduce the very evident leakage at the pulley holes but the net reduction is not likely to be a large percentage. Their application likely builds up the pressure slightly in the weight spaces and the leakage for the most part finds its way into the room through smaller and less noticeable openings between the window trim and plaster or at the baseboard. The proper place to stop the air leakage in good building construction would be at the points of entrance near the exterior surface of the wall.

Two 13-in. brick masonry wall sections and two wood frame wall sections were built to investigate frame leakages. Fig. 14 shows one of the masonry walls in the course of construction. Incompletely slushed walls were built in each case but mortar was slushed completely against the window frames. The room sides of the walls were plastered. Fig. 15 shows one of the two frame wall sections completely built and in position ready to close the machine for testing. The construction from outside to inside was bevel siding, building paper, sheathing, 2 x 4 studding, wood lath and plaster. Two variations were tested on each of these two walls. In one, the plaster ground stopped at the 2 x 4 framing for the window opening; this was termed *narrow plaster ground* test. In the other, the plaster ground bridged across from the 2 x 4 window framing to the window frame so as to offer greater resistance to air passage from the studding and weight spaces and was termed *wide plaster ground* test. The wall sections were built by average building mechanics to represent average building construction.

Table 5 shows the results of these tests for frame leakage for both types of wall. The average leakage obtained at 15 mph for the wood frame walls was 13.5 cu ft per hour referred to per foot sash perimeter. The infiltration through the five plain, not-locked windows with 3/32 in. crack and clearance was 124.5 cu ft and for these windows weatherstripped was 29.1 cu ft per hour per foot of sash perimeter at 15 mph. The aver-

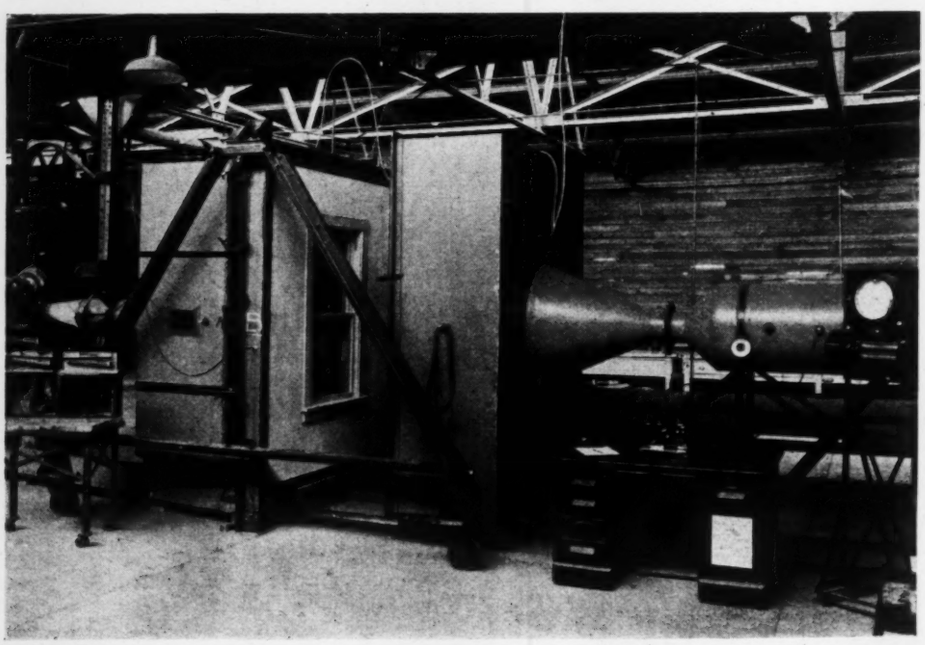

FIG. 15—WINDOW INSTALLED IN A WOOD FRAME WALL AND TESTED FOR FRAME LEAKAGE

TABLE 5—FRAME LEAKAGE IN FRAME AND MASONRY WALLS IN CUBIC FEET PER HOUR PER FOOT OF SASH PERIMETER

PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER	WIND VELOCITY MI/HR.	WOOD FRAME WALLS					MASONRY WALLS					
		NARROW PLASTER GROUND		WIDE PLASTER GROUND		AVE. OF FOUR TESTS	NOT CALKED			CALKED		
		WALL "A"	WALL "B"	WALL "A"	WALL "B"		WALL "A"	WALL "B"	AVE.	WALL "A"	WALL "B"	AVE.
0.012	5.0	3.2	2.6	2.5	2.3	2.7	6.2	1.9	4.1	0.8	0.4	0.6
0.027	7.5	5.9	5.0	4.5	4.5	5.0	10.2	3.8	7.0	1.5	0.9	1.2
0.048	10.0	9.0	8.0	6.6	7.0	7.7	14.3	6.3	10.3	2.3	1.5	1.9
0.075	12.5	12.3	11.0	8.8	9.7	10.5	18.4	9.2	13.8	2.9	2.2	2.6
0.108	15.0	16.0	14.4	11.0	12.7	13.5	22.7	12.3	17.5	3.6	2.9	3.3
0.147	17.5	20.3	18.2	13.7	16.0	17.1	27.0	15.6	21.3	4.3	3.7	4.0
0.192	20.0	24.9	22.2	16.3	19.5	20.7	31.5	19.1	25.3	5.0	4.4	4.7
0.243	22.5	29.7	26.3	19.1	23.1	24.6	36.3	22.7	29.5	5.7	5.0	5.4
0.300	25.0	34.9	30.8	22.2	27.0	28.7	41.6	26.3	34.0	6.4	5.6	6.0
0.431	30.0	46.4	40.5	28.9	35.5	37.8	53.1	33.5	43.3	8.0	6.4	7.2

age value for the nine plain, not-locked windows with 1/16 in. crack and 3/64 in. clearance which are considered to represent the fit of an average window was 35.6 cu ft and for the weatherstripped condition was 16.0 cu ft per hour per foot of sash perimeter at 15 mph. To these sash perimeter leakages should be added the 13.5 cu ft of frame leakage when applied to wood frame construction.

The average leakage through the frame in a masonry wall was found to be 17.5 cu ft per hour per foot of sash perimeter at 15 mph. Calking the frame joint resulted in reducing this leakage to 3.3 cu ft. When a perfect job of calking, which is easy to approach, is done, the frame leakage consists of air that enters into the brick work and travels to the window frame. This amounts to about 20 per cent of the uncalked frame leakage with the usual not completely slushed brick wall construction.

The particular frame leakage applying to the masonry wall, with the frame joint calked or not calked, should be added to the values of sash perimeter leakages to obtain the total leakage for the window opening.

Table 6 shows in the first line the average results for the plain and weatherstripped windows *D, H, I, M,* and *N* that were fitted to 3/32 in. crack and clearance. In addition, Table 6 shows the results of tests of seven windows termed *miscellaneous* of somewhat the same crack and clearance. These were tested as plain windows over the past several years and then weatherstripped. Only one test of each type is included in the miscellaneous tests although two different types of the same make are included in some cases. The thickness of the sash was 1⅜ in. in every case. The results shown in this table are those from the original tests for the plain windows. The results shown as for weatherstripped windows are from tests made during 1931. The cracks and clearances are the measured values taken in the past at the time of the original plain window tests. Since the cracks and clearances are only approximately the same as those used on the present series of five, and because the method of sealing is in question on a few of the plain window tests and because the cracks and clearances when tested as weatherstripped windows had probably changed from the original plain window values, it is considered best not to use an average obtainable from the inclusion of these so-called miscellaneous windows in Table 6. The average, allowing each window equal weight, however is given in this table for comparison to the average as determined in the present program on five windows of 3/32 in. crack and clearance. The average of the entire twelve is lower than the average of the five for both plain and weatherstripped conditions.

General Conclusions

Table 7 shows a summary of the frame leakages, the sash perimeter leakages and the sum of the two or the total window opening leakages for wood frame and for masonry wall constructions. In the case of masonry walls, only the total with the frame joint calked is given, since the benefit from calking is so obvious.

The Infiltration sections of A. S. H. V. E. GUIDES of 1930 and 1931 quote a leakage of 155.0 cu ft per hour per foot of sash perimeter at 15 mph. The crack is 1/16 in. and the clearance is 7/64 in. and the window not locked. The value for the weatherstripped window

TABLE 6—SUMMARY TABLE FOR INFILTRATION IN CUBIC FEET PER HOUR PER FOOT OF SASH PERIMETER THROUGH MISCELLANEOUS DOUBLE-HUNG WOOD WINDOWS WITH APPROXIMATELY 3/32 IN. CRACK AND CLEARANCE—NOT LOCKED

WINDOW	CRACK	CLEARANCE	PLAIN (6 MO. LATER)										WEATHERSTRIPPED									
			SASH PULLS POUNDS				MILES PER HOUR						SASH PULLS POUNDS				MILES PER HOUR					
			UPPER		LOWER								UPPER		LOWER							
			UP	DN.	UP	DN.	5	10	15	20	25	30	UP	DN.	UP	DN	5	10	15	20	25	30
AVE. FOR WINDOWS D, M, H, N, I	3/32	3/32	5	4	6	1	30.9	78.5	124.5	171.6	220.0	274.1	20	17	15	10	4.7	15.9	29.1	43.6	59.4	76.5
WINDOW O, WEATHERSTRIP	4	3/32	3/32	3	5	3	5	20.5	49.5	80.0	113.0	150.0	190.0	22	22	24	26	2.1	7.8	16.1	25.5	34.6	43.5
" P, "	5	1/8	3/32	7	4	4	2	27.5	74.8	126.0	175.5	223.0	266.0	18	16	10	4	2.4	10.0	20.5	32.5	45.0	57.0
" Q, "	6	1/8	3/32	-	-	-	-	14.5	50.0	81.0	116.0	147.0	175.0	8	6	10	7	5.1	18.0	34.4	52.0	68.6	85.3
" R, "	7	1/8	3/32	-	-	-	-	18.0	55.5	91.5	131.0	169.0	202.0	11	10	8	9	4.1	14.2	24.9	35.9	47.3	60.7
" S, "	8	7/64	3/32	2	1	9	2	25.0	58.0	91.5	122.5	152.9	181.0	15	11	25	20	5.0	14.8	25.2	36.0	48.2	62.0
" T, "	9	5/64	3/32	5	3	5	0	10.5	33.0	58.0	84.0	111.3	141.0	30	24	20	16	3.5	12.8	23.0	33.2	43.3	53.5
" U, "	10	5/64	3/32	5	5	8	0	8.7	27.3	49.3	72.5	97.3	123.5	34	34	40	28	2.6	9.7	19.7	31.2	42.8	54.0
WEIGHTED AVERAGE		3/32+	3/32	4	3	5	1	23.3	61.7	100.4	139.4	179.2	220.8	20	17	18	13	4.0	13.9	25.8	38.7	52.2	66.5

TABLE 7—SUMMARY TABLE OF INFILTRATION IN CUBIC FEET PER HOUR PER FOOT OF SASH PERIMETER THROUGH DOUBLE-HUNG WOOD WINDOWS—NOT LOCKED

PRESSURE DROP ACROSS WINDOW IN INCHES OF WATER	WIND VELOCITY ML/HR.	FRAME LEAKAGE			SASH PERIMETER LEAKAGE				FRAME LEAKAGE PLUS SASH PERIMETER LEAKAGE							
			MASONRY WALL		1/16" CR. & 3/64" CL.		3/32" CR. & 3/32" CL.		WITH WOOD FRAME WALL				WITH MASONRY WALL- CALKED			
		WOOD FRAME WALL	NOT CALKED	CALKED	PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED	1/16" CR. & 3/64" CL.		3/32" CR. & 3/64" CL.		1/16" CR. & 3/64" CL.		3/32" CR. & 3/32" CL.	
									PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED	PLAIN	WEATHER-STRIPPED
0.012	5.0	2.7	4.1	0.6	5.6	2.7	30.9	4.7	8.3	5.4	33.6	7.4	6.2	3.3	31.5	5.3
0.027	7.5	5.0	7.0	1.2	11.7	5.4	54.2	10.0	16.7	10.9	59.2	15.0	12.9	6.6	55.4	11.2
0.048	10.0	7.7	10.3	1.9	19.1	8.6	78.5	15.9	26.8	19.3	86.2	23.6	21.0	10.5	80.4	17.8
0.075	12.5	10.5	13.8	2.6	27.1	12.2	101.4	22.7	37.6	22.7	111.9	33.2	29.7	14.8	104.0	25.3
0.108	15.0	13.5	17.5	3.3	35.6	16.0	124.5	29.1	49.1	29.5	138.0	42.8	38.9	19.3	127.8	32.4
0.147	17.5	17.1	21.3	4.0	44.1	19.6	148.3	36.3	61.2	36.7	165.4	53.4	48.1	23.6	152.3	40.3
0.192	20.0	20.7	25.3	4.7	53.4	23.7	171.6	43.6	74.1	44.4	192.3	64.3	58.1	28.4	176.3	48.3
0.243	22.5	24.6	29.5	5.4	62.5	27.7	194.9	51.4	87.1	52.3	219.5	76.0	67.9	33.1	200.3	56.8
0.300	25.0	28.7	34.0	6.0	71.9	32.0	220.3	59.4	100.6	60.7	249.0	88.1	77.9	38.0	226.3	65.4
0.431	30.0	37.8	43.3	7.2	91.8	41.4	274.1	76.5	129.6	79.2	311.9	114.3	99.0	48.6	281.3	83.7

is given as 28.6 cu ft. A note states that these figures include the elsewhere leakage but not the frame leakage. These figures were arrived at by tests on windows with the seal made on the inside of the upper sash. The elsewhere leakage referred to as added is leakage that originated at the sash perimeter and entered the room through the pulley holes. These figures, then, are to be compared to the sash perimeter results of the tests on the five windows with a 3/32-in. crack and clearance of the present program. The average sash perimeter leakage of these windows, plain, was 124.5 cu ft and 29.1 cu ft for the weatherstripped window per foot of sash perimeter per hour at 15 mph.

The measurement of a large number of windows in actual buildings at this time indicates that the crack and clearance of the average window is considerably less than the foregoing GUIDE values of 1/16 in. crack and 7/64 in. clearance or the 3/32-in. crack and clearance of the series of five windows of the present program. According to these measurements and the tests on the nine windows of this program, the average window would have a 1/16-in. crack and a 3/64-in. clearance and as a plain not locked window would have a leakage of 35.6 cu ft per foot of sash perimeter per hour at 15 mph. When weatherstripped, its leakage on the average would be 16.0 cu ft. To these values should be added, when applied to wood frame construction, 13.5 cu ft of frame leakage, making a total of 29.5 cu ft per foot of sash perimeter per hour at 15 mph for the weatherstripped window. When applied to a calked masonry construction, the sash perimeter leakages should be increased by a 3.3 cu ft leakage making a total window opening leakage of 19.3 cu ft for the weatherstripped window. The plain window figures would be 49.1 cu ft for wood frame construction and 38.9 cu ft for masonry construction.

It is realized that the program reported in this paper is a laboratory program and that somewhat more representative figures might be obtained if a much larger number of windows were included in the tests. Furthermore, the effect of wear and weathering such as takes place in actual building constructions has not been considered. A valuable addition to the program as presented at this time might be made by the subjecting of the fourteen windows to wear and weathering. To accomplish this, they could be built into an enclosure that would be heated and on the outside subjected to weather conditions. The windows could be opened and closed a certain large number of times during a period of exposure of a year or more. The ease of opening and closing as measured by sash pulls under various weather conditions would check the belief of the authors that the weatherstrip installations on these fourteen windows represent average installation workmanship. The windows when removed from the exposure would be retested by the method followed in the tests of this paper to arrive at the leakage for a window as found in actual buildings.

Heat Transfer Through Building Materials

The U. S. Bureau of Standards has recently published the results of a study entitled, Heat Transfer Through Building Materials, by M. S. Van Dusen and J. L. Finck. See Bureau of Standards *Journal of Research*, March, 1931.

A method for measuring the heat transfer through large flat slabs, such as wall sections, is described in detail. The method requires no actual measurement of heat flow over large areas and no accounting for inevitable losses in undesired directions, except where lateral losses may occur by radiation, but consists in comparing the thermal resistance of an unknown panel with a standard, the resistance of which can be accurately determined by the hot-plate method, by which the measurement of heat flow is made under much more favorable conditions.

Results on 17 walls are given. The accuracy of the measurements is of the order of 5 per cent in the apparatus described. Greater accuracy than this can only be attained by better control of conditions, which becomes increasingly difficult as the temperature variation limits are narrowed. Such refinements would seem to be superfluous, however, in view of the degree of reproducibility of such structures as building walls.

In addition, results are given for five small panels, these observations being made with a hot plate 24 in. square. These smaller panels consisted of certain individual components of frame wall construction.

An analysis of the data permitted the calculation of the conductances and resistances of the various components entering into the construction of ordinary types of walls. These values are presented in tabular form, and their use permits the calculation of the resistance (or conductance) of any combination of the various components for which data are given.

Essential Elements for Determining Heating Plant Requirements

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OFTEN, that which seems to be a comparatively simple problem becomes involved and controversial because of failure to consider all of its related parts. Thus, the problem of determining the requirements for heating plants appears to be without any specific challenge. It is only a matter of supplying enough heat to maintain a comfortable living temperature and to take care of any heat losses from the building. Practically every one in the heating profession has a definite conception of the requirements of this problem as a whole; yet, when the integral parts are considered, there are wide differences of opinion. The purpose of this paper is to analyze the problem and to consider some of the causes for these differences.

Without quoting specific reference, free use has been made of the facts obtained by researches at such places as the A. S. H. V. E. Research Laboratory, the U. S. Bureau of Mines, the University of Illinois, the University of Wisconsin, and other institutions and research laboratories. An effort has been made to correlate the various parts of the problem and to arrive at a logical solution which would omit the unknown factors so often applied to heating plant design.

Insofar as the direct heat supply from the plant is concerned, there are at least four factors which have a direct, or, at least, an indirect bearing on the question. These are:

1. The air temperatures required within the space to be heated.
2. The heat utilization or the efficiency of the heat-distributing system used.
3. The unit of measure by which heaters are rated.
4. The method used for calculating heat losses from the building.

Other angles of the problem may present themselves, but it appears that these four questions are the bases for the greatest differences of opinion.

Air Temperatures Required

Everyone wants to be comfortable, and lack of comfort is probably the major cause for criticism of heating plants. The most direct relation which the heating plant bears to comfort is that of providing proper conditions so that the body can regulate its internal temperatures. The body is continually generating heat, and this heat must be dissipated at such a rate that the temperature of the body will be maintained within very narrow limits. This heat may be given directly to the surrounding air by convection and conduction, it may be absorbed by the evaporization of moisture, or it may be given off by radiation.

The relation of air conditions to the amount of heat given off directly to the air has been very carefully worked out by the A. S. H. V. E. Research Laboratory in cooperation with the U. S. Bureau of Mines. The relation of the amount of heat given off by radiation to the temperatures of the surrounding objects has also been given consideration. It is evident that if the surrounding conditions are such that the body radiates a large proportion of its heat, the amount given off directly to the air must be reduced, and, likewise, if the radiation factor is reduced, the amount given off by convection must be increased.

In practice, air having a temperature of 70 F has been considered as the average optimum condition to be aimed at by the heating plant designer. However, it can readily be seen that if the wall temperatures are very cold, such as might be the case if a person were surrounded largely by single glass windows on a sub-zero day, then it would be necessary that the air temperatures be materially higher in order to counterbalance the excess amount of heat or proportion of heat given off by direct radiation. On the other hand, if a person were surrounded entirely by inside walls which might be at a very high temperature, the air for comfort would then have to be a considerable amount below 70 F to satisfy the heat-dissipating requirements of the body.

While it is evident that a very wide range of temperatures may be required for comfort under different surrounding conditions and thus the comfort requirements may throw a variable factor on the heating plant, it is also evident that some practical, average condition must be assumed for use in the design of heating plants. This practical average has been almost universally accepted as 70 F.

Experience has shown that, with many types of heaters, there is a variation in room temperatures from floor to ceiling of as much as 15 deg to 25 deg. It will thus make a material difference as to how high from the floor the 70-F temperature is to be measured. Until recently, the most common point of measurement has been at 5 ft above the floor and not less than 3 ft from the outside wall, this being considered as the average breathing line. Recent investigators have considered this point too high, and 30 in. from the floor now seems to be an acceptable standard. If all radiators heated the room uniformly from top to bottom, the height from the floor at which temperatures were measured would make no difference either as to comfort or in the design of the heating plant. Unfortunately, however, many radiators do not have this uniform heating effect and, therefore, the load on the plant will be somewhat varied, depending upon the height assumed.

The effect of wall surface temperatures upon the

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radiant heat given off by the body should, undoubtedly, have further study in order to determine their effect on the comfortable living temperature, and, also, upon the requirements of the heating plant. The character of the wall surfaces also has something to do with the amount of heat given off by direct radiation from the body and, therefore, will affect the comfortable air temperatures required.

Heat Utilization

The efficiency with which heat is applied to the space to be heated has a direct bearing upon the heating-plant load. The requirements are to keep the body under comfortable living conditions as to temperatures of the air and of the surrounding objects. Since this condition is required at a distance of 30 in. from the floor, it is evident that excessively high temperatures at other parts of the room may result in a direct waste of heat, and, therefore, an overload on the heating plant. As has been previously pointed out, certain types of radiators will give temperatures from 15 deg to 25 deg higher at the ceiling than at the floor line. These high ceiling temperatures are useless, insofar as the comfortable conditions of the room are concerned, and for efficiency might well be avoided.

Take as an illustration a room with a 10-ft ceiling and with a 20-deg difference in temperature from floor to ceiling. With a straight line temperature variation and with 70 F at a point 30 in. from the floor, the floor line temperature will be 65 F and the ceiling temperature will be 85 F, or an average air temperature of 75 F which, with an outside temperature of zero would mean over 7 per cent additional heat loss through the outside walls. If the ceiling were exposed, the heat loss would be increased through the ceiling by slightly over 20 per cent as compared with the losses with ceiling air at a temperature of 70 F. The variations might be other than 20 deg, and the conditions surrounding the building, which would affect the loss by direct transmission and by infiltration, might be such as to increase the load on the plant anywhere from 0 up to 20 or more per cent, due to the inefficient application of the heat to the room.

This inefficiency in heat application is due mostly to the difference in temperatures from floor to ceiling and is caused largely by the type and location of the radiator or heating unit in the room. Any radiator which is so designed and so located that it will deliver air at warm temperatures to the top of the room will cause trouble, due to the stratification of the air, unless artificial means are used to break up this stratification and to cause the air to circulate. In cast-iron *radiation*, a low radiator of a given width has been found to give more uniform temperature distribution from floor to ceiling than a higher design of the same radiator. This is evidently due to the fact that the higher radiator delivers the air from the top at a higher velocity and a higher temperature. This air, in turn, reaches the ceiling at a temperature considerably above the average room temperature, remaining there to cause the uneven distribution. With the low radiator, the air is not thrown off with such high velocities or at such high temperatures, which gives it more of an opportunity to diffuse with the air in the room, and when it does reach the ceiling, insures that it will be at a lower temperature. Take the case of a wall type radiator placed under the window; experiments have shown that there is a very uniform temperature distribution from floor to ceiling. In this case, the warm air currents from the radiator pass up, meet, and diffuse with the cold air currents

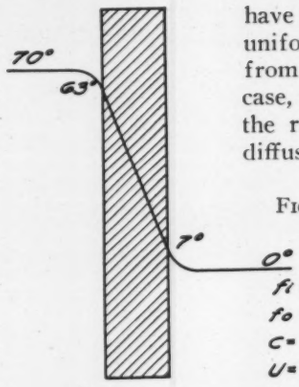

FIG. 1—MODERATELY INSULATED WALL

coming down from the window surface. These mixed air currents are at a much lower temperature than would be the case if the radiator were not placed under the window.

As far as the efficient distribution of convected air currents is concerned, the first step appears to be to prevent high temperature air from reaching the ceiling of the room. After it has once reached the ceiling, it cannot be brought down to the floor again by gravity current.

Besides the inefficiencies due to uneven temperature distribution, there may also be radiant heat losses. A part of the heat of most heating units is given off by direct radiation. This may vary from something over 30 per cent in ordinary cast-iron *radiation* down to very small percentages in some of the convector types of heating units. The effectiveness of radiant heat depends upon what happens to it after it leaves the radiator. Radiant heat does not affect the air directly but must strike some object which will absorb it and transmit it to the air by conduction and convection. If radiant heat strikes inside wall surfaces or objects, it should, in general, be effective in supplying the heat requirements for the building. If, on the other hand, radiant heat strikes outside wall surfaces which are very cold, such as glass, its effectiveness may be greatly reduced.

As an illustration of what may happen to radiant heat when coming in contact with outside wall surfaces, refer to Figs. 1 and 2. Fig. 1 represents a reasonably well insulated wall, the overall heat transmission coefficient (U) of which is 0.15, and Fig. 2 represents a single glass window with an overall heat transmission coefficient of 0.75 for still air. If, for Fig. 1, the inside air temperature is assumed to be 70 F and the outside, zero, with reasonably still air on both sides of the wall, the surface coefficients may be taken as 1.5 and the temperatures of the wall surfaces would be 63 F and 7 F, respectively. If radiant heat were to strike this wall on the inner

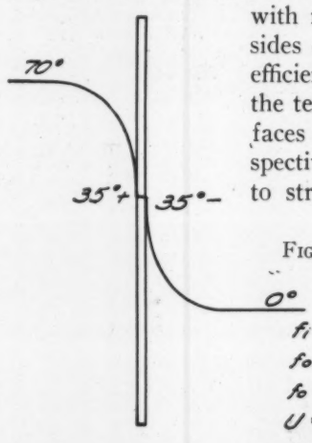

FIG. 2—SINGLE GLASS WINDOW

surface, the effect would be to build up the inner surface temperature and increase the heat flow through the wall to that extent. As the wall temperature approached 70 F, the amount of heat given to the wall by convection from the air would be decreased. With an extreme condition, in which most of the heat were given off by radiation, the wall temperature might reach 70 F. In this case, the heat loss through the wall would be increased by 11 per cent. This would be a maximum condition and would represent a radiator in which practically all of the heat was given off by direct radiation and all radiator surfaces exposed to outside wall surfaces, a condition which would be very unlikely. It is thus evident that radiant heat is not a serious factor when we consider walls which are well insulated.

Next, consider the wall represented by Fig. 2, in which case the glass has a very low internal heat resistance, the greater part of the resistance to heat flow being on the two surfaces. With air at 70 F and zero, respectively, on the two sides of the glass, it is reasonable to assume that the average temperature of the glass surfaces will be 35 F. In this case, if radiant heat were to strike and be absorbed by the inner surface, the temperature would rise and the heat loss through the glass would be increased substantially proportional to this rise in temperature. If an extreme case is assumed in which the inner surface is brought up to 70 F by radiant heat, the heat loss through the glass would be nearly doubled.

A further condition which might occur is that of a wind velocity on the outside of the surface, such as to increase the outer surface coefficient from 1.5 to 5.0 or more. If, under this condition, the inside surface temperature were raised to 70 F, the total heat loss through the thin wall might be increased by six or more times the original amount.

The exact amount of radiant heat which might be lost through thin walls depends upon several factors, such as surface coefficients, wind velocities, internal heat resistance of wall, and amount of radiant surface exposed to outside walls. In order that radiant heat may be effective, it should be received by objects in the room or by inside walls and not by outside walls which have very low thermal resistance. Moreover, it should be received by objects which are at a low elevation in the room in order to maintain the temperature distribution as low in the room as possible.

From the foregoing, it is evident that heat utilization may be an important factor in considering the load on the heating plant, but it is further apparent that the effectiveness of both convected and radiant heat is somewhat under the control of the designer. With the proper consideration for the laws governing heat distribution from heating units, a very high heat utilization factor should be obtained.

Unit of Measure for Heaters

The question of heat utilization brings up that of a proper method to measure heating units. It is evident that the effectiveness of various types of heaters is different, due to the fact that they distribute their heat in the room by different methods. If this difference in effectiveness is not taken into account in rating or measuring the output of the unit, it will place an uncertain factor on the final load required for the heating plant. It is true that this difference may lie well within the factor of safety assumed in the design; yet it exists and should not escape attention. The question has received a great deal of consideration and discussion in the past and is especially pertinent when it comes to the selection of test methods and of a test room for the rating of heating units.

In general, there are two types of rooms which have been used for this purpose. The first is one in which the wall temperatures are substantially the same as the temperature of the air in the test room, and the heat dissipation from the room takes place by an interchange of air to the space surrounding the test room. The second type of test room is one in which the heat given off by the radiator is absorbed directly by the walls of the test room, these walls being cooled by refrigerated air on the outside or by means of spraying cold water on the outer surfaces of the walls.

In the first type of room, the convection currents from the radiator are somewhat disturbed, and the heat distribution from top to bottom of the room is changed, due to the air circulation into and out of the test room. If the test room is so arranged that there is a free circulation of air out of the top and in at the bottom, it is quite possible that a radiator which in practice would result in a large temperature difference from floor to ceiling would show only a nominal difference in the test room, due to the fact that the hot air is freely discharged at the top and thus not allowed to build up in temperature. These convection currents may not be such as seriously to affect the heat output from the radiator, but they may affect very materially the temperature distribution as measured throughout the different elevations of the room.

A second point which should be given consideration in this type of room is that of the absorption of radiant energy by the walls. Since all surrounding walls are substantially at the same temperature as the air in the room, they would virtually be inside walls and the radiant energy might be more effective than would be the case if the radiator were used in an actual room with wall surfaces below the temperatures of the surrounding air. In the majority of cases, the magnitude of this error is probably small, due to the fact that, in most radiators, only a small proportion of the heat is given off by direct radiation, and of this, only a small proportion strikes the outside cold surfaces directly from the radiator. Furthermore, as has been shown in the discussion of Fig. 1, if these cold surfaces have any appreciable amount of internal resistance, the effect of the radiant heat losses is only a small proportion of the total radiant heat.

Taking everything into consideration, it is evident that this type of test room disturbs the temperature distribution factor very materially, but, in most cases, does not have any great effect upon the total heat output from the radiator. The room is not well adapted to test a radiator for heat distribution, but is a convenient form of test room for determining the total heat output by the condensation method. It is flexible and can be adjusted for different capacities.

Considering the second type of test room, this can be made to approach a practical condition, providing the walls are cooled in such a manner that their inner surface temperatures will equal those met in practice. For radi-

ators giving off the greater portion of their heat by convection, it would appear to make no difference as to the internal heat resistance of the walls or as to the methods used in cooling them to bring the internal or inside surface temperatures to the proper degree. As long as these temperatures are properly maintained, the heat conducted from the air in the room to the surface of the wall by convection currents should be the same as met with under practical conditions.

If radiant heat is considered, the method of cooling the walls, and the amount of internal resistance in the test room walls may have a decided effect upon the efficiency or heating effect of the radiator. Take, for instance, the very thin metal wall which is cooled by circulating water over the outside surfaces to maintain an inside surface temperature at 63 F. In this case, there is an unlimited supply of 63 F water, which will maintain the internal surface temperature regardless of the amount of heat given to the wall. Thus, if radiant heat strikes this wall, it is absorbed by it without raising the inside temperature, and, therefore, does not retard the heat given to the wall by convection, which means that all radiant heat is virtually lost, without in any way affecting the air temperature within the room. If, in this type of test room, a radiator which gives off 30 per cent of its heat by radiation were tested and so placed that half of the radiant heat reached the cold walls, it would mean a 15 per cent loss without any heating effect to the air in the room. If this same radiant heat struck a test wall in which there was a large internal heat-resisting factor, it would serve to raise the inside surface temperature, and, therefore, would not be a direct loss. The direct loss would be reduced as the internal resistance of the wall was increased. If a convection type of heater were to be tested in this room, the only function of the test room cold walls would be to absorb the heat from the air, as the air temperature would be raised within the radiator itself and would not depend upon the radiant heat being intercepted by the wall surfaces and returned to the air in the room. It is conceivable that a radiating unit might be so designed as to emit practically all of its heat by direct radiation, in which case the total output from the radiator might be entirely ineffective if the radiant heat should strike one of the thin metallic walls cooled to a pre-determined inside surface temperature by an unlimited amount of circulating water on the outside surface. This radiant heat could not be so lost if the walls contained a practical amount of internal heat resistance. In this case, the surface temperatures of the wall would be increased until the heat was transmitted to the air in the room.

From the foregoing, it is evident that the type of test room is very important if radiators of different design and characteristics are to be tested for comparison. If heat distribution is to be one of the factors obtained from the test, the method of dissipating the heat by an interchange of air to the exterior of the room may lead to appreciable errors in the heating effect as indicated from the test. If the total heat supplied by the radiator is the only factor regardless of its effectiveness in heating the room, then the first type of testing room apparently has advantages and the heat output may be measured by the condensation method.

If heat distribution, as well as heat supply, is to be measured, the second type of room or the one in which the wall temperatures are maintained at some practical degree below those of the air for at least a part of the walls seems to have the advantage. If the units under test are designed to emit the greater part of their heat by convection currents, there should be no material difference as to the method of cooling the walls or as to the internal resistance in the cool walls. On the other hand, if the heat units to be tested give off a part of their heat by radiation and these radiators are so placed that this radiant heat strikes the cool walls, it is then essential that the walls be designed with an average amount of internal heat resistance, in order that the radiant heat which strikes the walls may build up the surface temperatures and obtain the same effectiveness as would be the case under practical conditions.

It is evident that a test room cannot be built which will meet every condition in practice, but one should be constructed which will meet average conditions. The first type of test room is flexible, for testing different sizes of heaters in the same room. It does not seem to be well adapted to testing heaters for temperature distribution, but is reasonably consistent for determining the output by the condensation method. The second type of room is not readily adapted to different sizes of heaters. For convection-type heaters, the walls may be of any construction and cooled by any method so long as the inner surfaces present a practical surface at the proper temperatures. For radiant heaters, the cold walls must have a practical amount of internal heat resistance in order that the radiant heat may be effective. With these precautions, it may be used to advantage for determining heat distribution on the heat output by the condensation method.

The next question which has always been a disturbing factor in considering the problem of rating radiators is that of a proper method to measure the output of the radiator even after a test room has been selected. The most common and easily applied method has been to measure the output by the condensation method; that is, the radiators have been tested, the total condensation under given conditions noted, and the rating determined by converting this condensation into British thermal units. This method is easily applied and gives the total amount of steam required of the heating plant, but does not show accurately the results which may be expected from a given type of heater.

Since there is admittedly a marked difference in the heat distribution from various types of heaters, it seems that this should at least be recorded by an acceptable test method. For this purpose, the first type of test room does not seem to be the best selection, and especially not unless very great care is given to the method of interchanging the air in the room. If this type of room is to be used, the interchange should be effected in such manner as not to disturb the temperature gradient from top to bottom of the room as set up by the natural convection currents of the radiator itself.

Taking everything into consideration, the best type of test room would appear to be one in which the proper amount of outside wall surface is cooled and the walls are constructed with an average amount of internal heat resistance in order to give the radiant heat an opportunity to build up the surface temperatures and thus to be

effective as a heating medium. Further than this, it would seem that the room temperature should be measured at a height of 30 in. from the floor line, at which point 68 F should be maintained, and the other temperatures from top to bottom should be recorded as a matter of showing the effectiveness of heat distribution from the particular radiator under test. There is undoubtedly a need for further research work to determine the proper type of test room and the proper test methods in order to take into consideration all of the important factors in connection with a heating unit.

Calculation of Heat Losses from Buildings

The fourth and most important factor in determining heating plant requirements is that of calculating the heat losses from a building. In fact, this may be considered as the main part of the problem and as including in part the other three subdivisions as previously discussed. The major factors to be considered in this part of the problem are as follows:

1. Air conditions required within the building.
2. Outside weather conditions to be expected.
3. Thermal properties of the walls used in the construction.
4. Air leakage or infiltration through the various parts of the building.
5. The quality of workmanship which may be obtained during construction.

The complete information regarding these five factors has not, in the past, been at the disposal of the designer. This has led to much uncertainty, and, in many cases, the use of large and variable factors of safety. The lack of information on certain parts of the problem has served to cast a suspicion and doubt on data obtained for certain other parts and upon methods of calculation which appear scientifically correct, but for which certain necessary factors or coefficients have not been available.

There are two general paths or ways by which heat may escape from a building.

1. By direct transmission through the walls of the structure.
2. By infiltration or an interchange of air from the inside to the outside of the building.

The laws governing the flow of heat by direct transmission have had a great deal of study, and it is now possible to calculate with accuracy the heat flow through most types of walls under conditions of constant temperature difference between the air on the two sides of the wall. Under this condition, the transmission is governed by the temperatures and velocities of the air on the two sides of the wall, by the internal heat resistance of the wall, and, to a certain extent, by the temperatures of the surrounding objects. When these conditions remain constant, it is comparatively easy to determine the demand on the heating plant. If, however, there are sudden changes in the air temperatures, the demand on the heating plant may not immediately follow these changes, due to the heat capacity of the wall.

Consider two walls, one built of brick 24 in. thick, and the other built of cork 1½ in. thick. These walls have practically the same thermal resistance. The first wall has a heat capacity of approximately 40 Btu per square foot of wall per degree drop in temperature, and the second wall has about 0.4 Btu. of heat capacity under the same condition.

If a drop of 10 deg in the outside temperature is assumed, the additional heat flow through each wall after uniform conditions are reached would be $0.2 \times 10 = 2.0$ Btu per hour. The total heat storage of the brick wall after the changed conditions were reached would be 40×5 or 200 Btu less than before the temperature drop. It is evident that a wall of this type would be very slow in reaching normal conditions of heat flow and that the 200 Btu would act as a heat storage to relieve the maximum demand on the plant. The cork wall, on the other hand, would have only 2 Btu of reserve capacity and could not cause any such delay or lag in the heating plant requirements. A glass window would cause practically no lag in the maximum requirements.

This heat lag is caused by the stored-up heat in the wall and depends upon its heat capacity. The effectiveness of this heat capacity depends upon the conductivity of the wall. If the conductivity is low, it will last for a longer period of time than it will if the conductivity is high. The extent to which this heat capacity will affect the maximum requirements of the heating plant will depend not only upon the capacity and the conductivity of the wall, but upon the length of time during which the minimum outside temperatures prevail. It would seem that it can best be taken into account by a thorough study of the thermal properties of walls, together with an analysis of weather bureau records. From this analysis, outside temperatures may be selected for determining the maximum heating plant requirements which will apply to various types of construction. The necessity of considering thermal capacity of the structure as an element in calculating maximum heating plant requirements has been recognized by the American Society of Heating and Ventilating Engineers and an extended research program is under way at the Research Laboratory.

Wind velocity affects the direct heat transmission through a wall by changing the surface coefficient. From Table 1 it is evident that the effect is entirely different for walls of different internal heat resistances. No single factor can be obtained which will fit the various wall constructions and correlate wind velocity to temperature differences.

The logical procedure for determining heating plant requirements for walls subject to direct transmission appears to be as follows:

1. Select the inside air temperature to be maintained.
2. Consider the outside weather conditions to be expected. This includes the minimum temperature, length of time these temperatures prevail, maximum wind velocities, and direction to be expected.
3. Consider the thermal properties of the structure and select proper outside temperatures and conductivity coefficients to be used in the design of heating plant. From these data, calculate by well-known formulae the total amount of heat by direct transmission. Such a procedure needs no factor of safety or exposure factor, providing the proper care is used in selecting the coefficients.

Table 1—Comparison Between Walls in Figs. 1 and 2 for Still Air and a 35-mph Wind Velocity

Wall	f_o Still Air	f_o 35-Mile Wind	Temp. Drop Air to Air	U Still Air	U 35-Mile Wind	Percent Increase in U
Fig. 1	1.5	8.0	70	0.15	0.163	8.67
Fig. 2	1.5	8.0	70	0.75	1.26	68.0

The loss of heat by infiltration is an entirely different loss than that by direct transmission and requires a different treatment. The amount depends upon the size of the cracks around the windows and doors, the construction of the walls, the wind velocity and direction on the outside of the building, and, to a certain extent, the exposure. The losses due to infiltration have in the past been the most uncertain part of the problem, and have often been very high, especially under heavy wind conditions. This uncertainty has led to the use of exposure factors which, on the windward side of the building, have at times run as high as 40 per cent. Such exposure factors can only be justified by the lack of knowledge which has existed in regard to these losses. They are an admission that something has been omitted in the design, and this has usually been the effect of high wind velocities on increasing the air infiltration. Wind velocities have been assumed which are too low and the amount of air leakage into the building has been under-estimated. With data which are now available, it is possible to determine more accurately the losses by infiltration, and, therefore, there is less need to use exposure factors.

Conclusions

In conclusion, consider the five specific parts of the problem as outlined on page 601 for determining the heat loss from buildings:

1. The air conditions required within the building relate largely to air temperatures unless ventilation is required in which the heat loss through this channel should be considered as a separate part. With heating units which give good distribution of temperature from floor to ceiling, it should be reasonable to assume 70 F as the average temperature of the air within the room. If, however, heating units are to be used which give variations of 20 or more degrees from floor to ceiling, it will then be necessary to use higher average wall temperatures and especially higher temperatures for ceilings which are exposed.

2. The outside air conditions to be expected constitute one of the most difficult parts of the problem. This includes the minimum temperatures, wind velocities, the effect of sunshine on the building, temperature changes, and exposure. The minimum temperature and the direction and velocity of the wind may be obtained from the Weather Bureau records for most cases. The most difficult part of the problem, however, is to determine the maximum combined effect of temperature and wind velocity for any one period. As previously pointed out, wind velocity affects heat transmission through the wall by changing the outside surface coefficient, which may make a rather high percentage of differences for poorly insulated structures. Wind velocity has a direct effect on infiltration, and for high wind velocities, the infiltration losses of poorly constructed buildings may be greater than the direct heat transmission losses. Complete information on this subject should give the maximum wind velocity and direction to be found for any temperature condition during the heating season. It would then be possible for the designer to determine the maximum requirements for the building. The effects of sunshine and temperature change may be of less importance in determining the maximum load on the heating plant than they are for determining the maximum load on the cooling plants for summer conditions. They are, however, very important factors in determining maximum requirements. They must be considered in conjunction with heat capacity of the structure as previously explained.

3. The heat transmission, through various types of construction, has received a great deal of attention and a great deal of data are available, covering different types of wall construction. It is true that the greater part of the coefficients have been considered from the point of constant temperature differences, whereas the maximum load on the heating plant may be greatly affected by the heat capacity of the wall and the duration of low temperatures to be expected. This part of the problem needs more thought and research. A consistent study of the weather bureau records is also necessary in order to determine the proper outside temperatures to use with walls of different thermal properties.

4. The air infiltration losses have probably caused more failures and have been more difficult to determine than any other part of the problem, except, perhaps, the outside weather conditions to be expected. This is natural, as there are so many factors involved. With proper construction, however, these losses may be greatly reduced, and, from data now available, may be very closely determined. The data which are now most needed may be obtained from a comprehensive analysis of the weather bureau records to find the wind velocity and direction which may expected at different temperature ranges. Losses can then be accurately determined, but any failure to determine them cannot be scientifically taken care of by an exposure factor, as there is no logical reason for applying such an exposure factor on one side of the building more than on any other, excepting for the reason that proper coefficients were not selected.

5. The quality of construction which may be expected, may play a very important part in the heat losses. It will affect both the direct transmission losses and the air leakage losses. The most logical method of allowing for this seems to be to take it into consideration when selecting the transmission coefficients and infiltration factors. If any factor of uncertainty were to be applied to the calculations, this would seem to be the most logical point of application.

Wind velocities and temperature gradients have two distinct effects on the amount of heat loss from a building. Wind velocity affects transmission by changing the outside surface coefficient of the wall. For well insulated walls, it will have but little effect, while for thin, highly conductive walls, it may have a very appreciable effect. Temperature differences affect the transmission in direct proportion to the differences. There is no relation between the two which will fit all cases.

Take as an illustration the two walls of Figs. 1 and 2. For reasonably still air, the outside surface coefficients are 1.5, and for wind velocities of 35 mph, they will be about 8.0. Under these conditions, the conductivities for a temperature drop of 70 F to zero, would be as shown in Table 1. From these results, it is evident that no factor will correlate wind velocities and temperature differences.

The factors which at present seem to require more information in the problem of calculating maximum heating plant requirements are: *first,* a more comprehensive study of weather bureau records; *second,* a definite recognition of the heat capacity of walls.

Sizing Pipes and Orifices for Gravity Hot Water Heating Systems

This paper is the result of research conducted at the Engineering Experiment Station of Texas Agricultural and Mechanical College in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

By Elmer G. Smith[1], College Station, Texas
MEMBER

A SET of tables intended for use in designing gravity circulation hot water heating systems for residences and other buildings of small or moderate size, was presented by the writer at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in Philadelphia, Pa., January, 1930[2]. After a careful analysis of these tables, the Technical Advisory Committee on Pipe Sizes for Heating Systems (1930-31) came to the conclusion that the pipe sizes obtained by means of them would result in systems too costly for general use. It was therefore decided to continue the investigation in order to develop another set of tables by which smaller and more economical pipe sizes could be obtained. It also seemed desirable to revise the orifice formula so that it would give more accurate results for small "radiators." In order to reduce pipe sizes without reducing radiator efficiencies it was necessary to reduce safety factors, and to reduce safety factors without jeopardizing circulation required much more accurate calculations than those upon which the first tables were based.

Every effort has been put forth to make the tables presented in this paper as practical as possible. The present study covers bare mains. A study of insulated mains has not yet been made, but it seems probable that these data will give satisfactory results if the insulation has a low efficiency. It is not necessarily true that a system having covered mains is better than a system having bare or partly covered mains. The bare mains not only contribute considerable heat to the basement but also by warming the basement ceiling they help to heat the first story and make it more comfortable.

Where basement "radiation" is required it is essential that a ceiling type of "radiator" be used and connected in a definite manner as explained later in this paper.

Definitions

In order to avoid confusion, the following terms which are used in this paper, have the definitions indicated:

Riser. A vertical pipe.

Boiler Uptake. A vertical pipe leading up from a boiler outlet to a supply main or header.

Boiler Return. A vertical pipe leading down from a return main to a boiler inlet or header.

Supply Mains. The horizontal piping leading from the boiler uptake to the supply "radiator" or riser branches.

Return Mains. The horizontal piping leading from the return "radiator" or riser branches to the boiler returns.

Supply Riser Branch. The piping which connects a supply main to a supply riser.

Return Riser Branch. The piping which connects a return riser to a return main.

Supply "Radiator" Branch. The piping which connects a supply main or riser to a "radiator."

Return "Radiator" Branch. The piping which connects a "radiator" to a return main or riser.

Expansion Pipe. The piping which connects the system to the expansion tank.

Overflow. The piping which connects the top of the expansion tank to the sewer or other means of drainage.

Circulating Pipe. The piping which connects the expansion tank to the bottom of the boiler or to some part of the return piping.

Vent Pipe. The pipe at the top of the overflow which is open to the atmosphere in order to prevent syphoning the system.

Although the tables contained in this paper were calculated primarily for use in connection with systems having open expansion tanks, they can also be used for closed systems. An open expansion tank system designed according to these tables and operated at its maximum capacity should show an average temperature differential of about 30 deg for the "radiators," being higher than this for "radiators" close to the boiler and lower for "radiators" far from the boiler. Under these conditions the temperature differential at the boiler should be about 40 deg in an average installation. When a system is being operated at less than its maximum capacity the temperature differentials will all be lower than the calculated values but their relative values will be about the same. On the other hand, if a system is put under pressure and made to operate at a capacity greater than would be possible for a similar open system, the temperature differentials will all be somewhat higher than for the open system.

From this it can be seen that if the pipe and orifice sizes for a closed system are selected on a basis of the "radiation" actually used, the system will operate at a higher temperature differential than would an open system, and the blow-off valve will have to be set to main-

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[1] Assistant Professor of Physics, Texas Agricultural and Mechanical College.

[2] See Pipe and Orifice Sizes for Small Gravity Circulation Hot Water Heating Systems, by Elmer G. Smith [JOURNAL of the Society (Heating, Piping and Air Conditioning), February, 1930].

For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1931.

tain a pressure of at least 15 lb per square inch at the top of the highest "radiator" when the system is being operated at its maximum capacity.

If, on the other hand, the pipe and orifice sizes are selected as if for an open system and only the sizes of the "radiators" reduced, the system will operate at about the same temperature differential as an open system. In this case the blow-off valve can be set to maintain a pressure of 10 lb per square inch at the top of the highest "radiator" when the system is being operated at its maximum capacity.

The first method makes the cost of installation less and gives smaller heat losses in the basement, which makes it more suitable for a system having bare mains. However, the second method is less likely to cause trouble by leaking because of the lower pressure and lower maximum temperature of the supply piping.

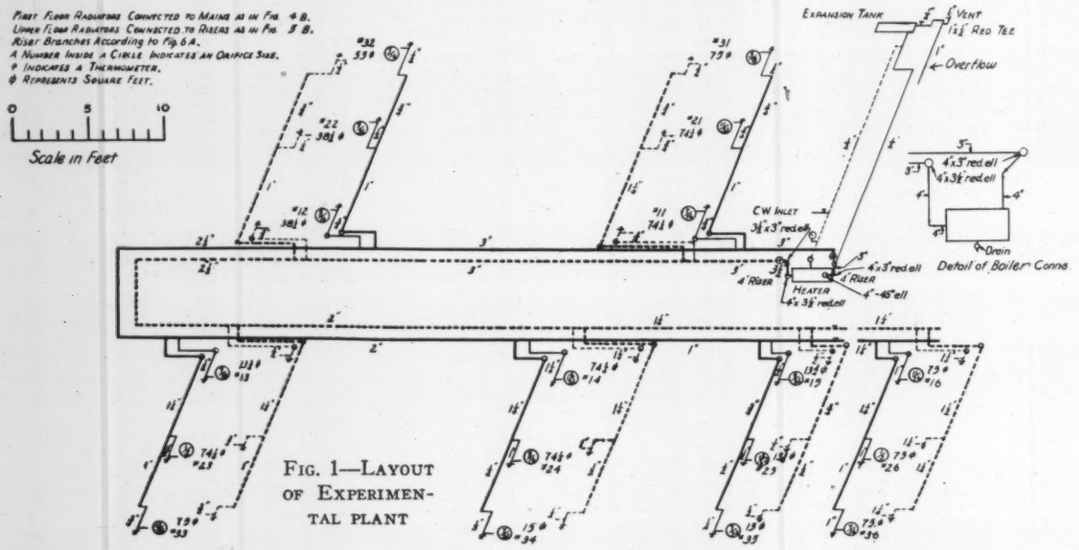

First Floor Radiators Connected to Mains as in Fig. 4 B.
Upper Floor Radiators Connected to Risers as in Fig. 5 B.
Riser Branches According to Fig. 6 A.
A Number Inside a Circle Indicates an Orifice Size.
Indicates a Thermometer.
Represents Square Feet.
Scale in Feet
Expansion Tank
Vent
Overflow
C.W. Inlet
Drain
Detail of Boiler Conns.
Heater

FIG. 1—LAYOUT OF EXPERIMENTAL PLANT

Results of Tests on a Three-Story Experimental Plant

The platforms and "radiators" used to test these tables were the same ones that were used to test the tables submitted in the original paper on this subject.[2] The converter used to heat the water for the system was altered in order to reduce its internal resistance. Even after the alterations, the internal resistance of this converter may have been higher than that of an ordinary "boiler" having the same size outlets and inlets, but the difference was not enough to be serious. The sizes of the pipes and orifices were changed to correspond with those given in the new tables. The layout of the experimental plant is shown in Fig. 1. "Radiators" No. 16, No. 26, and No. 36 are not a part of the system as originally designed and were used principally to determine the effect of overloading and to permit one large "radiator" (No. 16) to be substituted for the three small ones on the end of the line (No. 15, No. 25, and No. 35). There was no insulation on any part of the mains, risers, riser branches, or "radiator" branches during any of these tests.

The first test on this system was run with the "radiators" standing on the scaffold with both front and back exposed and the mains running less than one foot above a cold floor. There was no flooring or ceiling above the mains such as there would have been in an ordinary building. Consequently, the cooling of the water in the mains was greater than it would have been under average conditions. In the remaining tests the "radiators" and risers were backed on one side by wall board (See Fig. 2) in order to approximate the effects of a wall, and the mains were partly covered by wall board as illustrated in Fig. 3. The wall board was applied loosely with numerous holes around risers and scaffolding, and one end was left open so that the enclosure would not become too hot. This may not have closely simulated average basement conditions but it at least provided a warm ceiling with a cool floor and only a moderate amount of convection. The results of these tests are given in Table 1.

In the first test, the "radiators" and pipes were exposed much more than they would have been in an ordinary building. The hottest "radiator" was No. 23 with an average temperature of 167½ F, and the coolest "radiators" were Nos. 25 and 35, each of which had an average temperature of 155 F, a difference of 12½ deg. However, the heat dissipated per square foot of equivalent "radiation" was practically the same for all three of these "radiators" because No. 25 and No. 35 were each only five-section "radiators" while No. 23 was a 27-section "radiator." The "boiler" was intended to operate on a 40-deg differential and on the first floor the temperature differentials were intended to be about 37½ deg in section 1, about 34½ deg in section 2, about 29½ deg in section 3, and about 22½ deg in section 4. Also, the "radiator" temperature differentials were intended to decrease as the heights of the "radiators" above the "boiler" increased. Most of the observed temperature differentials were reasonably close to their calculated values, but the temperature differential of the "boiler" was considerably too large. This may have

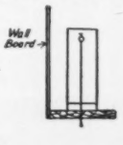


FIG. 2—SUPPLY END OF RADIATOR SHOWING WALL BOARD BACKING

[2] See Pipe and Orifice Sizes for Small Gravity Circulation Hot Water Heating Systems, by Elmer G. Smith [JOURNAL of the Society (Heating, Piping and Air Conditioning), February, 1930].

TABLE 1—RESULTS OF TESTS ON EXPERIMENTAL PLANT

First Test: No basement ceiling or walls. Expansion tank open. Readings taken 3 hours after starting.

Radiator Number	Boiler	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Supply Temperature	190½	186	182	176	179	170½	184½	180½	181	178	167	181	177	179	169	163
Return Temperature	144	145	149	145	152½	147	150	146	154	155	143	141	143½	151	150½	147
Average Temperature	167¾	165½	165½	160½	165¾	158¾	167¼	163¼	167½	166½	155	161	160¼	165	159¾	155
Temperature Differential	56½	41	33	32	26½	23½	34½	34½	27	23	24	40	33½	28	18½	16

Air Temperatures: First floor 59 F, Second floor 60.5 F, Third floor about 61 F.

Second Test: Basement ceiling and walls simulated. Expansion tank open. Readings taken 2½ hours after starting.

Radiator Number	Boiler	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Supply Temperature	188	186	182	178	179	172	185½	181	182	179½	169	181	177	179	170	164
Return Temperature	146	146	152	152	156	152	153	151	158	159	151	145	140	153	159	149
Average Temperature	167	166	167	165	167½	162	169¼	166	170	169¼	160	163	158½	166	164½	156½
Temperature Differential	42	40	30	26	23	20	32½	30	24	20½	18	36	37	26	11	15

Air Temperatures: First floor 71 F., Second floor 69.5 F.

Third Test: Basement ceiling and walls simulated. Expansion tank closed. Maximum pressure 30 lb per square inch at gage, or 20 lb per square inch at the expansion tank. Readings taken 3½ hours after starting.

Radiator Number	Boiler	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Supply Temperature	256	249	245	240	242½	234	247	244	245	242	229	245	243	242½	231	223
Return Temperature	201½	194	204½	208	216	208½	198½	203	215	215	202	211½	207	212	213	204½
Average Temperature	228¾	221½	224¾	224	229¼	221¼	222¾	222	230	228½	216½	228¼	225	227½	222	213¾
Temperature Differential	55½	55	41½	32	26½	26	48½	41	30	27	27	34½	36	30½	18	18½

Air Temperatures: Basement 79 F, First floor 69.5 F, Second floor 70.5 F, Third floor 76 F.

Fourth Test: Basement ceiling and walls simulated. Expansion tank open. Readings taken 3¼ hours after starting.

Radiator Number	Boiler	11	12	13	14	16	21	22	23	24	31	32	33	34
Supply Temperature	204	200	197	192½	193½	189	199	194½	194½	192	196	193	193½	186
Return Temperature	161½	161	168	164	171	163½	166	161	170½	171	166	166	166½	168
Average Temperature	182¾	180½	182½	178¼	182¼	176¼	182½	177¾	182½	181½	181	179½	180	177
Temperature Differential	42½	39	29	28½	22½	25½	33	33½	24	21	30	27	27	18

Air Temperatures: Basement 86 F, First floor 84 F, Second floor 82 F, Third floor 86 F.

Fifth Test: A continuation of fourth test. Expansion tank closed. Maximum pressure 25 lb per square inch at the gage, or 15 lb per square inch at the expansion tank. Readings taken 3¼ hours after closing expansion tank.

Radiator Number	Boiler	11	12	13	14	16	21	22	23	24	31	32	33	34
Supply Temperature	251½	244	241	236	238	233	243	239	239	237	238	237	236	227
Return Temperature	199	191	198	203	212	202	197	196	209½	211	207½	204	206	209½
Average Temperature	225¼	217½	219½	219½	225	217½	220	217½	224¼	224	222¾	220½	221	217¼
Temperature Differential	52½	53	43	33	26	31	46	43	29½	26	31½	33	30	17½

Air Temperatures: Basement 84 F, First floor 77 F, Second floor 76.5 F, Third floor 83.5 F.

been partly due to the fact that the "boiler" resistance was a little too high but it seems probable that it was due principally to excessive heat losses from the mains which are usually located just under the cellar ceiling where it is hot, instead of just above a floor where it is cold.

The "radiator" temperature differentials ranged from 41 deg for No. 11 to 16 deg for No. 35, becoming less as the distance from the "boiler" increased. This was the result of a deliberate effort to secure uniform average "radiator" temperatures.

In the second test the "radiators" were shielded on one side by wall board as shown in Fig. 2, and the mains were shielded in a manner somewhat similar as shown in Fig. 3. In this test, all the temperature differentials were satisfactorily close to the calculated values. The temperature differential of No. 34 was too low, but since it was approximately correct in all the other tests, it seems probable that there was an error in reading one of the thermometers.

In both the first and second tests, it required about 10 min for the water to travel the length of the mains, a distance of 98 ft. This corresponds to a speed of about 10 ft per minute. Sixteen minutes from the starting time every "radiator" had begun to receive hot water. Twenty-eight minutes from the starting time all the "radiators" were filled with hot water and by forty-five minutes from the starting time the average temperature of every "radiator" was above 140 F. The average temperatures of some of them were above 150 F at this time.

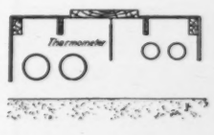


FIG. 3—SECTIONAL VIEW OF MAINS SHOWING WALL BOARD ENCLOSURE

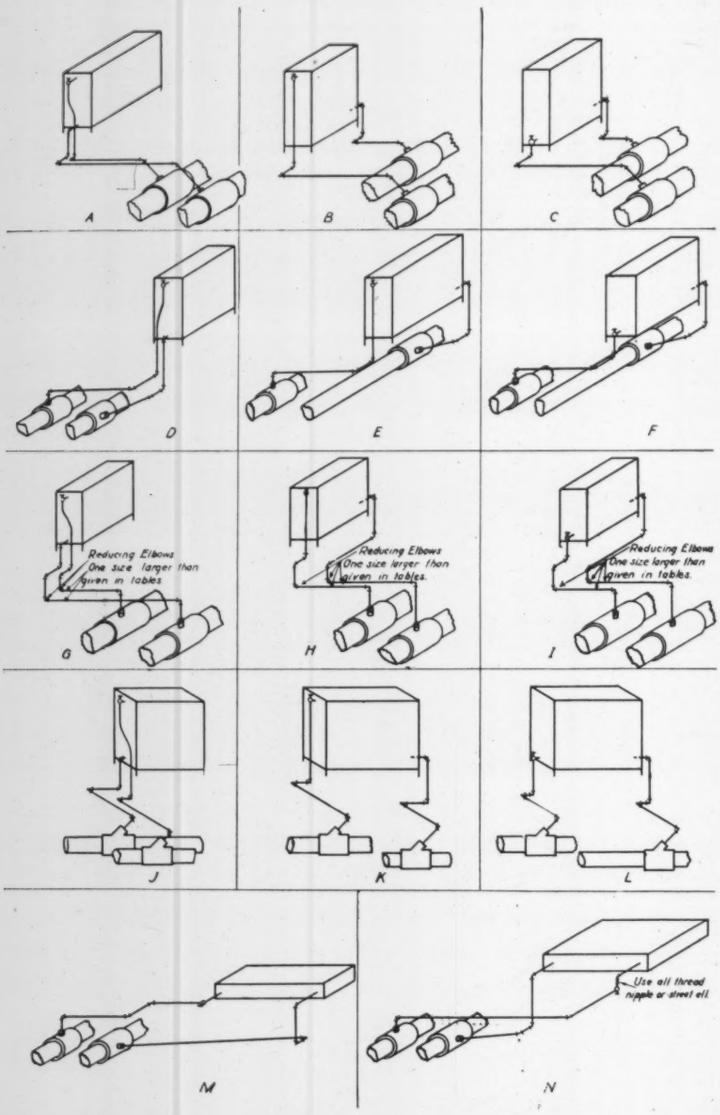
The third test was run with the expansion tank closed and the steam pressure on the converter raised from 18 lb per square inch to 65 lb per square inch. No

blow-off valve was provided but whenever the water pressure went above 30 lb per square inch, gage, the drain was opened until this pressure had fallen to 28 lb per square inch. The rate of heating up was more rapid than in the first test and at the end of an hour the average temperature of the system was 200 F. At this time the water was leaving the heater at 223 F, and returning at 169 F. When the system reached equilibrium, the hottest "radiator" was No. 23 with an average temperature of 230 F. The coolest "radiator" was No. 35 with an average temperature of 213¾ F, a difference of 16¼ degrees. However, the coolest "radiator" having more than five sections was No. 11, which had an average temperature of 221½ F, so that when a correction is made for the higher efficiencies of small "radiators," it is found that the actual maximum difference in rate of heat emission corresponds to a temperature difference of 8½ deg which is very small for such high "radiator" temperatures.

A small "radiator" located on the end of a line is always likely to circulate poorly and should be avoided whenever possible. The experimental plant was erected in this manner in order to make certain that the tables would give reasonably good results even when the arrangement of the "radiators" was such as to make good circulation difficult.

In order to find out what might be expected from a more normal installation the risers carrying "radiators" No. 15, No. 25 and No. 35 were cut out and "radiator" No. 16 was connected in their place. Since the total rating of the disconnected "radiators" was only 42½ sq ft, while the rating of No. 16 was 75 sq ft, this constituted a slight overload upon the mains, but since the overload amounted to less than half the rating of the last first-floor "radiator" (No. 16), no serious consequences were anticipated. The fourth test was then run with this set-up. When equilibrium had been established, the hottest "radiators" were No. 12, No. 21, and No. 23, each of which had an average temperature of 182¼ F. The coolest was No. 16 with an average temperature of 176¼ F, a difference of 6 deg. Apparently the only effect of the overload was to increase the temperature differential of "radiator" No. 16 from the calculated 22½ deg to 25½ deg and thereby to lower its average temperature by 1½ deg.

As soon as the fourth test was completed the fifth test was started by simply closing the expansion tank vent and raising the steam pressure on the converter to 65 lb per square inch. When the system had again come to equilibrium, it was found that the hottest "radiator" was No. 14 with an average temperature of 225 F. The coolest were Nos. 11, 16 and 22, each of which had an average temperature of 217½ F. The temperature difference between the hottest "radiator" and the coolest was therefore 7½ deg. During this test the pressure at the gage was never higher than 25 lb per square inch, which means that the pressure at the expansion tank never went above 15 lb per square inch.

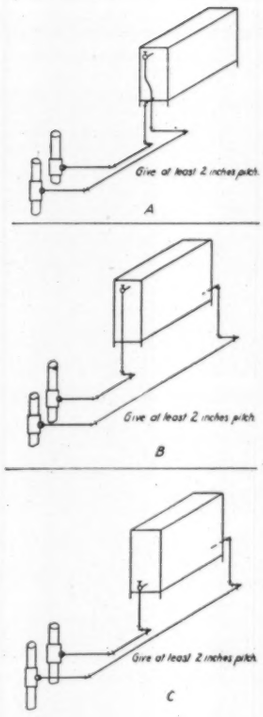


FIGS. 4A TO 4N—FIRST-FLOOR RADIATOR CONNECTIONS TO WHICH TABLES ARE INTENDED TO APPLY

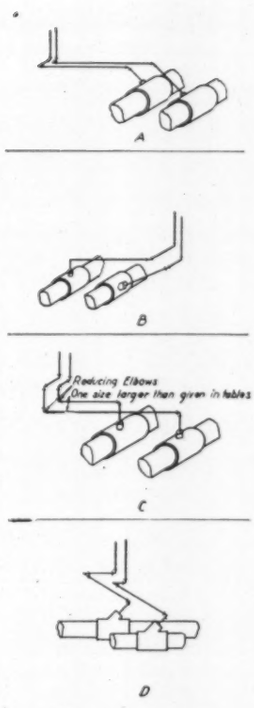
Types of Piping to Which Tables Apply

The types of first floor "radiator" branches to which these tables are intended to apply are illustrated in Figs. 4*A* to 4*L*, inclusive. The types of branch shown in Figs. 4*A*, *D*, *G* and *J* probably give somewhat better circulation through the radiator than any of the others and those shown in Figs. 4*B*, *E*, *H* and *K* probably give slightly better "radiator" circulation (higher average radiator temperature) than those shown at *C*, *F*, *I* and *L*. Also, the high supply connection places the valve in a convenient position where it is not necessary to stoop to operate it. On the other hand, the low supply connection takes longer to become air bound than the high supply connection.

Two possible methods of connecting basement "radiators" are shown at Figs. 4*M* and 4*N*. These connections contain more elbows than are necessary but if fewer elbows were to be used for basement "radiators" than for first floor "radiators," it would complicate the tables and this did not seem to be advisable. For basement "radiators" it is always advisable



FIGS. 5A TO 5C—METHODS OF CONNECTING RADIATORS ABOVE FIRST FLOOR



FIGS. 6A TO 6D—METHODS OF RUNNING LATERALS

to keep the supply radiator branch above the return radiator branch. The connections shown at Fig. 4*M* are better from the standpoint of circulation because the supply branch is higher and shorter, but the valve is easier to reach in Fig. 4*N*. The connections to the mains are usually made as illustrated in Figs. 4*A*, *B* and *C*, but the method shown in Figs. 4*D*, *E* and *F* is just as good except that the return branch is not quite as flexible.

In using the connections shown in Figs. 4*D*, *E* and *F*, it is important to note that the supply main must be farther from the "radiator" than the return main in order that the connection to the supply main may be vertical and to the return main horizontal. Close to the "boiler," this precaution may be disregarded, but in connecting "radiators" near the ends of the mains, it is important. The connections shown in Figs. 4*G*, *H* and *I* are extremely flexible and should be used whenever the "radiator" riser passes through a tight slot or hole that will not permit it to move freely. This is especially important near an elbow in the mains because at such places the mains move toward and away from the wall as well as parallel to it. The connections shown in Figs. 4*J*, *K*, *L* give better circulation through the mains than any of the others because the velocity heads are not entirely lost as the water enters and emerges from the "radiator" branches, but the large *Y* connections may be difficult to obtain and the connection as a whole is not quite as flexible as some of the others.

It is expected that the length of a "radiator" branch, measured horizontally from the mains to the "radiator" will normally be about 4 ft. If this distance is more than 8 ft it is advisable to use the orifice and pipe sizes that are ordinarily required for the next larger "radiator."

The methods of connecting the "radiators" on floors higher than the first are shown in Figs. 5*A*, 5*B* and 5*C*. Any other type of connection containing approximately the same number of elbows and having the same vertical flexibility is equally satisfactory. It is expected that on the average a "radiator" will be about 4 ft from its risers. As in the case of the first-floor "radiators," it is good practice to use the orifice and pipe size ordinarily required for the next larger "radiator" when the "radiator" is more than 8 ft from the risers.

It is usually advisable to anchor or support a riser at the point where the second floor "radiator" branches enter it. This allows for a considerable amount of horizontal flexibility at the base of the riser where it joins the riser branch and at the same time prevents any appreciable vertical motion at this point.

The riser branches should be installed according to Figs. 6*A*, 6*B*, 6*C* or 6*D*. When the riser branch is larger than the riser the change should be made by means of a reducing elbow at the foot of the riser. That is, all the fittings must be in the branch and none in the riser. This is important.

The mains may be run either side by side as in Fig. 7*A*, or the supply main may be placed directly above the return main, as in Fig. 7*B*. The side-by-side arrangement gives more head room but the supply-main-above arrangement makes it easier to connect branches and also insures somewhat more stable circulation. Either arrangement, however, is entirely satisfactory, provided the remainder of the installation is properly made. The return main should never be run above the supply main as this might cause instability.

In calculating the capacities of the mains, allowance was made for four elbows in each main. This number should not be exceeded unless compensated by the use of larger pipe. Elbows located so near the end of the

FIGS. 7A AND 7B—ARRANGEMENT OF SUPPLY AND RETURN MAINS

line that they affect only one "radiator" can usually be neglected, but elbows near the "boiler" are very important and should be avoided whenever possible. If it is necessary to locate more than one elbow in each main near the "boiler," the best plan is to use larger fittings. One size larger will permit two elbows instead of one and two sizes larger will permit four. It should be remembered that a tee is equivalent to 2.2 elbows. Tees should therefore be avoided whenever possible. The elbows or reducing elbows at the tops of the "boiler" uptake and "boiler" return are included in the "boiler" connections and need not be considered when counting the elbows in the mains.

The best method of connecting the "boiler" uptakes and returns is shown at Fig. 8*A*. The connections shown in Figs. 8*B* and 8*C* also give good results. The connections shown at Fig. 8*D* are acceptable provided the line contains not more than three elbows per main, none of which should be close to the heater. Since the mains are usually smaller than the main heater risers, the elbows at the tops of the "boiler" uptakes and re-

FIGS. 8A TO 8D—METHODS OF CONNECTING THE MAIN HEATER RISERS

turns will usually be reducing elbows. There is, of course, considerable latitude in the location of the cold-water feed and the drain but there is probably a little less danger of air binding if the cold-water feed as well as the drain is located near the bottom of the "boiler" because the air driven out of the water by the heat will

TABLE 2—CAPACITIES OF ORIFICES AND RADIATOR BRANCHES. IF THE RADIATOR BRANCH IS MORE THAN 8 FEET LONG, MEASURED HORIZONTALLY FROM THE MAINS OR RISERS TO THE RADIATOR, USE AN ORIFICE ONE SIZE LARGER THAN SPECIFIED

| DRILL SIZE | PIPE SIZE | SECTION 4 — HEIGHT ABOVE MIDPOINT OF BOILER IN FEET — SECTION 4 | | |

have more time to collect in large bubbles before leaving the "boiler" and will therefore be more likely to be caught by the expansion pipe and kept out of the "radiators."

The expansion pipe connections shown in Figs. 8*A*, 8*B*, 8*C* and 8*D* are good because they are located so as to divert from the "radiators" a large portion of the air that is driven out of the water in the boiler. Frequently the expansion tank is located in such a cold place that it must be protected from freezing by circulation and in this case a return line or circulating pipe from the expansion tank, preferably to the bottom of the "boiler," is required.

If the expansion tank is located at the top of a riser and connected to both the supply and return it should be treated as a "radiator" and orificed. The risers should be sized to carry their load of "radiators" plus another "radiator" as large as the expansion tank orifice would carry. If this is not done the pressure difference between the risers may be so reduced that poor circulation will result. The expansion tank connections shown in Figs. 9*A* and 9*B* are good because if the overflow is

TABLE 2—CAPACITIES OF ORIFICES AND RADIATOR BRANCHES. IF THE RADIATOR BRANCH IS MORE THAN 8 FEET LONG, MEASURED HORIZONTALLY FROM THE MAINS OR RISERS TO THE RADIATOR, USE AN ORIFICE ONE SIZE LARGER THAN SPECIFIED

PIPE SIZE	DRILL SIZE	SECTION 3 — HEIGHT ABOVE MIDPOINT OF BOILER IN FEET — SECTION 3																															DRILL SIZE	PIPE SIZE
		CONNECT TO MAINS ACCORDING TO FIGS. 4A-4N.													CONNECT TO RISERS ACCORDING TO FIGS. 5A-5C.																				
		3-3½	3½-4	4-4½	4½-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17	17-18	18-20	20-22	22-24	24-26	26-28	28-30	30-33	33-36	36-40	40-43	43-46	46-50	50-55	55-60		
½	7/64							2.0	2.1	2.3	2.4	2.5	2.7	2.3	2.4	2.5	2.6	2.7	2.9	3.1	3.2	3.4	3.5	3.7	3.9	4.1	4.4	4.5	4.7	4.9	5.2	5.4	7/64	½
	33						2.0	2.1	2.3	2.5	2.6	2.7	2.9	2.5	2.6	2.7	2.8	2.9	3.2	3.3	3.5	3.7	3.9	4.0	4.3	4.5	4.7	4.9	5.1	5.4	5.7	5.9	33	
	32					2.0	2.2	2.4	2.6	2.8	3.0	3.1	3.2	2.8	2.9	3.0	3.2	3.3	3.5	3.8	3.9	4.1	4.3	4.5	4.8	5.0	5.3	5.5	5.7	6.0	6.3	6.6	32	
	31				2.0	2.2	2.5	2.7	3.0	3.1	3.3	3.5	3.7	3.2	3.3	3.5	3.6	3.7	4.0	4.3	4.5	4.7	4.9	5.1	5.4	5.7	6.0	6.3	6.5	6.9	7.2	7.6	31	
	1/8				2.1	2.4	2.8	3.0	3.2	3.4	3.6	3.8	4.0	3.4	3.5	3.8	3.9	4.1	4.4	4.6	4.9	5.1	5.4	5.6	5.9	6.2	6.6	6.9	7.1	7.5	7.8	8.2	1/8	
	30		2.0	2.3	2.5	2.9	3.2	3.5	3.6	4.0	4.3	4.5	4.8	4.1	4.3	4.5	4.7	4.9	5.2	5.5	5.8	6.1	6.4	6.6	7.0	7.4	7.8	8.1	8.5	8.9	9.3	9.8	30	
	29	2.0	2.2	2.5	2.8	3.2	3.6	3.9	4.2	4.5	4.8	4.9	5.3	4.5	4.7	4.9	5.2	5.3	5.7	6.1	6.4	6.7	7.0	7.3	7.7	8.1	8.6	9.0	9.3	9.8	10.3	10.8	29	
	9/64	2.2	2.6	2.9	3.1	3.6	4.1	4.4	4.8	5.1	5.3	5.6	6.0	5.1	5.4	5.7	5.9	6.1	6.5	6.9	7.3	7.7	8.0	8.3	8.8	9.3	9.8	10.2	10.6	11.1	11.7	12.3	9/64	
	26	2.4	2.8	3.1	3.4	4.0	4.4	4.9	5.3	5.6	5.9	6.1	6.6	5.6	5.9	6.2	6.5	6.7	7.2	7.6	8.0	8.4	8.8	9.2	9.7	10.2	10.8	11.3	11.7	12.2	12.9	13.5	26	
	24	2.6	3.0	3.4	3.7	4.3	4.8	5.2	5.7	6.0	6.4	6.7	7.1	6.1	6.4	6.7	7.0	7.2	7.7	8.2	8.7	9.1	9.5	9.9	10.5	11.0	11.7	12.2	12.6	13.2	13.9	14.6	24	
	5/32	2.9	3.3	3.7	4.0	4.7	5.3	5.8	6.2	6.6	7.0	7.4	7.7	6.6	6.8	7.4	7.6	7.8	8.4	8.9	9.3	9.9	10.3	10.7	11.4	11.9	12.7	13.2	13.7	14.3	15.1	15.8	5/32	
	20	3.1	3.6	4.0	4.4	5.1	5.6	6.2	6.7	7.1	7.5	8.0	8.3	7.2	7.5	7.9	8.2	8.5	9.2	9.7	10.2	10.7	11.2	11.7	12.3	13.0	13.7	14.3	14.9	15.6	16.4	17.2	20	
	19	3.4	3.9	4.4	4.8	5.6	6.2	6.8	7.3	7.8	8.3	8.8	9.1	7.9	8.2	8.6	9.0	9.3	10.0	10.6	11.2	11.8	12.3	12.8	13.5	14.2	15.1	15.7	16.3	17.1	17.9	18.8	19	
	11/64	3.8	4.4	5.0	5.4	6.3	7.0	7.6	8.1	8.7	9.3	9.7	10.2	8.9	9.3	9.7	10.1	10.5	11.3	12.0	12.6	13.2	13.8	14.4	15.2	16.0	17.0	17.7	18.3	19.2	20.2	21.2	11/64	
	15	4.2	4.9	5.5	6.0	6.9	7.7	8.4	9.1	9.7	10.2	10.8	11.3	9.8	10.3	10.7	11.3	11.7	12.5	13.3	14.0	14.7	15.3	16.0	16.9	17.8	18.8	19.6	20.3	21.3	22.4	23.5	15	
	3/16	4.7	5.5	6.1	6.7	7.7	8.6	9.4	10.2	10.8	11.4	11.9	12.6	10.9	11.5	11.9	12.6	13.0	14.0	14.8	15.6	16.4	17.1	17.8	18.8	19.8	21.0	21.9	22.7	23.8	25.0	26.3	3/16	
	9	5.2	5.9	6.6	7.3	8.4	9.5	10.3	11.0	11.6	12.3	12.9	13.6	11.9	12.5	13.1	13.6	14.2	15.2	16.2	17.1	18.0	18.6	19.4	20.5	21.4	22.7	23.7	24.8	25.9	27.3	28.6	9	
	13/64	5.5	6.4	7.1	7.8	9.0	10.0	10.9	11.7	12.4	13.2	14.0	14.8	12.7	13.3	13.9	14.6	15.2	16.2	17.2	18.2	19.0	19.9	20.7	21.9	23.0	24.4	25.4	26.4	27.6	29.1	30.3	13/64	
	4	6.1	7.1	7.9	8.6	9.9	11.0	12.1	13.0	13.7	14.5	15.2	15.9	14.1	14.8	15.5	16.2	16.8	18.0	19.1	20.2	21.3	22.0	22.9	24.2	25.5	27.0	28.2	29.2	30.6	32.2	33.8	4	
	7/32	7.1	8.1	9.1	10.0	11.5	12.7	13.9	15.0	15.8	16.7	17.5	18.3	16.4	17.2	18.0	18.8	19.4	20.8	22.1	23.3	24.5	25.6	26.6	28.1	29.6	31.4	32.7	33.9	35.5	37.4	39.2	7/32	
	15/64	8.1	9.3	10.4	11.4	13.2	14.4	15.7	16.8	17.8	18.8	19.6	20.5	18.6	19.5	20.4	21.3	22.1	23.6	25.1	26.5	27.8	29.0	30.2	31.9	33.6	35.6	37.1	38.5	40.3	42.5	44.5	15/64	
	1/4	9.0	10.4	11.6	12.7	14.5	16.0	17.4	18.5	19.6	20.6	21.6	22.6	20.8	21.8	22.8	23.8	24.7	26.5	28.1	29.6	31.0	32.4	33.8	35.7	37.5	39.8	41.5	43.0	45.0	47.4	49.8	1/4	
	17/64	10.0	11.5	12.8	14.0	15.9	17.6	19.0	20.4	21.5	22.6	23.6	24.5	22.8	23.9	25.0	26.2	27.1	29.1	30.8	32.5	34.1	35.7	37.1	39.3	41.3	43.8	45.7	47.3	49.6	52.2	54.8	17/64	
	9/32	11.6	13.4	14.8	16.2	18.4	20.2	21.8	23.2	24.4	25.6	26.6	27.7	26.5	27.8	29.1	30.4	31.6	33.8	35.9	37.9	39.6	41.5	43.2	45.6	48.0	50.9	53.0	55.0	57.6	60.7	63.7	9/32	
	5/16	13.0	15.0	16.6	18.0	20.3	22.4	24.2	25.7	27.0	28.1	29.0	30.4	29.6	31.0	32.5	33.9	35.1	37.7	40.1	42.2	44.2	46.3	48.0	50.9	53.5	56.7	59.2	61.3	64.3	67.7	71.0	5/16	
	11/32	14.0	16.3	18.0	19.7	22.4	24.5	26.2	27.6	28.8	29.8	30.4	31.9	32.0	33.6	35.1	36.6	38.1	40.7	43.4	45.6	47.8	50.0	52.0	55.0	57.8	61.3	63.9	66.3	69.5	73.1	76.7	11/32	
	3/8	15.5	17.8	19.9	19.8	22.9	25.3	27.5	29.8	31.4	33.4	34.4	36.5	35.6	37.3	39.0	40.8	42.3	45.9	48.2	50.8	53.1	55.5	57.9	61.2	64.3	68.2	71.0	73.7	77.3	81.4	85.5	3/8	
¾	3/16				21.7	25.1	27.7	30.2	32.3	34.1	35.9	37.9	39.4																					3/16	¾
	13/64	16.8	19.4	21.6	23.6	27.0	30.0	32.6	35.2	37.1	39.1	41.0	42.7	38.7	40.6	42.5	44.3	46.0	49.3	52.4	55.1	57.8	60.5	62.9	66.5	70.0	74.2	77.3	80.2	84.0	88.5	92.9	13/64	
	7/32	19.4	22.4	25.0	27.1	31.2	34.4	37.2	40.0	41.9	44.5	46.2	48.2	44.5	46.7	48.9	51.0	52.9	56.6	60.2	63.5	66.5	69.6	72.4	76.6	80.5	85.0	89.0	92.3	96.7	100.6	106.8	7/32	
	3/8	21.7	25.1	27.8	30.2	34.7	38.2	41.4	44.0	46.3	48.8	51.0	53.1	49.8	52.3	54.7	57.0	59.2	63.5	67.5	71.0	74.5	77.8	81.0	85.7	88.5	95.5	98.0	101.5	106.5	113.4	119.5	3/8	
	15/32	23.7	27.4	30.4	32.9	37.5	41.6	44.8	47.8	50.3	52.7	54.8	56.8	53.6	57.1	59.7	62.3	64.6	69.4	73.6	77.5	81.3	85.0	88.6	93.6	98.4	104.3	108.7	112.8	118.1	124.3	130.5	15/32	
	7/16	26.9	31.2	34.4	37.2	42.5	46.5	50.0	53.3	56.0	58.4	60.5	62.8	61.6	64.6	67.6	70.6	73.3	78.4	83.4	87.8	92.0	95.4	100.3	106.0	111.5	118.1	123.1	127.8	133.7	140.8	147.5	7/16	
	½	29.3	33.9	37.8	38.8	44.6	49.2	53.9	58.0	61.6	64.9	68.2	71.1	67.5	70.8	74.0	77.5	80.2	86.0	91.4	96.2	101.0	105.4	109.8	116.1	122.0	129.4	135.0	140.0	146.5	154.0	161.8	½	
–	17/64				41.3	47.5	52.6	57.2	61.0	65.0	68.6	71.8	75.2																					17/64	–
	9/16	32.9	38.0	42.4	46.0	53.1	58.7	63.8	68.5	72.0	76.4	79.5	83.1	75.6	79.5	83.0	86.7	90.0	96.5	102.6	107.9	113.2	118.2	123.0	130.0	136.9	145.0	151.2	156.9	164.2	173.9	181.5	9/16	
	19/32	36.2	41.9	46.5	50.9	58.3	64.2	69.8	74.8	78.7	83.2	86.8	90.3	83.4	87.4	91.5	95.5	99.0	106.2	113.0	118.8	125.4	130.1	135.5	143.1	150.6	159.8	166.3	172.6	180.8	190.0	199.5	19/32	
	½	42.1	48.1	54.0	59.0	67.2	73.9	80.0	85.0	89.4	94.6	97.5	102.1	96.5	101.2	106.0	110.9	114.9	123.0	131.0	137.9	144.2	151.0	157.0	166.0	174.6	185.2	193.0	200.0	210.0	220.8	231.9	½	
	9/16	46.8	54.1	60.0	65.5	74.0	81.4	87.9	93.2	98.2	103.0	107.0	110.9	106.0	111.4	117.5	121.8	127.1	136.4	145.0	152.5	160.0	167.1	174.1	182.6	193.4	205.3	214.0	222.0				9/16	
	5/8	51.5	59.6	66.5	65.8	76.0	85.2	91.7	99.4	104.8	111.0	116.8	122.9	119.0	125.0	130.5	136.0	141.5	151.2	161.1	169.5	177.9	186.0	193.6	204.6	215.0							5/8	
1¼	17/32				72.7	83.8	92.8	101.3	108.5	115.2	122.0	127.0	132.3																					17/32	1¼
	9/16	56.3	65.2	72.8	79.3	91.8	101.2	110.2	118.0	125.2	132.5	138.5	144.5	130.1	136.2	143.0	149.0	154.8	165.8	176.0	185.5	194.2	203.2	210.0	223.8								9/16	
	19/32	61.4	70.5	77.6	84.6	97.5	109.2	118.9	127.5	134.6	142.5	149.0	156.2	140.8	147.1	154.8	159.6	167.5	179.5	190.5	201.0	210.3	219.2										19/32	
	5/8	69.5	80.6	88.8	97.6	112.3	124.0	134.6	143.5	152.0	161.0	167.5	175.3	160.8	168.1	176.1	183.6	191.0	204.0	217.0													5/8	
	11/16	77.5	89.6	98.2	108.2	124.0	136.8	149.8	159.0	168.2	175.7	183.5	191.0	177.8	186.5	195.0	203.2	211.0															11/16	
	¾	88.5	102.5	114.2	123.8	141.9	155.8	168.1	178.5	188.6	197.0	206.0	214.5	203.9	214.0	222.4																	¾	
	13/16	99.3	114.9	128.0	125.6	144.7	160.0	174.0	186.0	198.0	209.3	219.3																					13/16	
1½	¾				140.0	160.7	179.0	195.0	209.0	221.0																							¾	1½
	15/16	108.3	125.7	139.9	152.2	174.8	192.6	209.0																									15/16	
	7/8	123.5	143.4	158.2	173.0	197.0	216.8																										7/8	
	1	132.7	154.2	171.4	186.7	213.2																											1	
	1⅛	145.3	168.1	186.0	202.0																												1⅛	
	1⅜	162.2	187.3	206.0																													1⅜	
**	1⅛	164.7	214.0																														1⅛	**
	1⅜	212.2																															1⅜	

** RADIATOR BRANCHES CONSIST OF 1½" RISERS WITH 2" CONNECTIONS TO THE MAINS. THE REDUCING ELBOWS LOOK UP

FIGS. 9A AND 9B—EXPANSION TANK CONNECTIONS

nearly straight and vertical the reducing tee prevents any possibility of overflow even when the vent pipe is very short.

General Procedure in Designing a System

For convenience the process of designing a gravity hot water system may be divided into ten steps, as follows:

Step 1.

Select the "radiator" sizes so that each "radiator" shall be required to deliver the same number of Btu per equivalent square foot. The maximum amount of heat required from each "radiator" should not exceed 150 Btu per equivalent square foot for open systems. For closed systems the maximum should not exceed 216 Btu per equivalent square foot. "Radiators" should be

TABLE 2—CAPACITIES OF ORIFICES AND RADIATOR BRANCHES. IF THE RADIATOR BRANCH IS MORE THAN 8 FEET LONG, MEASURED HORIZONTALLY FROM THE MAINS OR RISERS TO THE RADIATOR, USE AN ORIFICE ONE SIZE LARGER THAN SPECIFIED

| PIPE SIZE | DRILL SIZE | SECTION 2 — HEIGHT ABOVE MIDPOINT OF BOILER IN FEET — SECTION 2 | | |

located under windows whenever possible. A "radiator" under each window is the best arrangement when comfort takes precedence over cost.

Step 2.

Make the layout using only such arrangements of piping as are illustrated in Figs. 4 to 9, but do not attempt to select pipe sizes at this point. As far as possible place a large "radiator" on the end of each line.

Step 3.

Check the layout to make sure that the mains do not contain more than four elbows each, neglecting the one between the last two "radiator" or riser branches on the line, if such a one occurs.

Step 4.

Check the layout to see that there is not more than one elbow per main near the "boiler" unless the plan of the building is such as to make more than one absolutely necessary.

Step 5.

Measure the length of the longest line from the "boiler" uptake to the farthest end. Divide this length by seven and arrange the line in four sections, as follows:

Section 4: the 1/7 of the line farthest from the heater.
Section 3: the 2/7 of the line next to section 4.
Section 2: the 2/7 of the line next to section 3.

TABLE 2—CAPACITIES OF ORIFICES AND RADIATOR BRANCHES. IF THE RADIATOR BRANCH IS MORE THAN 8 FEET LONG, MEASURED HORIZONTALLY FROM THE MAINS OR RISERS TO THE RADIATOR, USE AN ORIFICE ONE SIZE LARGER THAN SPECIFIED

PIPE SIZE	DRILL SIZE	SECTION 1 — HEIGHT ABOVE MIDPOINT OF BOILER IN FEET — CONNECT TO MAINS ACCORDING TO FIGS. 4A-4N.											
		3-3½	3½-4	4-4½	4½-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13
½	7/64		2.1	2.3	2.4	2.7	2.9	3.1	3.3	3.6	3.7	3.9	4.0
	33	2.1	2.3	2.5	2.6	2.9	3.2	3.4	3.6	3.8	4.0	4.2	4.4
	32	2.4	2.6	2.8	2.9	3.3	3.6	3.8	4.1	4.4	4.6	4.8	4.9
	31	2.7	3.0	3.2	3.4	3.7	4.0	4.3	4.6	4.9	5.1	5.4	5.6
	1/8	3.0	3.2	3.4	3.7	4.0	4.4	4.7	5.0	5.4	5.6	5.9	6.1
	30	3.5	3.8	4.1	4.3	4.8	5.2	5.6	6.0	6.4	6.7	7.0	7.2
	29	3.9	4.2	4.7	4.8	5.3	5.8	6.2	6.6	7.0	7.3	7.7	8.0
	9/64	4.4	4.8	5.1	5.5	6.0	6.6	7.1	7.5	8.0	8.3	8.7	9.1
	26	4.9	5.3	5.5	6.0	6.6	7.2	7.7	8.3	8.7	9.2	9.6	10.0
	24	5.3	5.7	6.1	6.5	7.2	7.8	8.4	8.9	9.4	9.9	10.3	10.7
	5/32	5.7	6.2	6.6	7.0	7.8	8.5	9.1	9.7	10.2	10.7	11.2	11.7
	20	6.2	6.7	7.2	7.6	8.4	9.2	9.8	10.5	11.1	11.6	12.2	12.7
	19	6.8	7.4	7.9	8.4	9.3	10.1	10.8	11.5	12.2	12.7	13.3	13.8
	11/64	7.7	8.3	8.7	9.4	10.4	11.3	12.1	12.9	13.6	14.3	14.9	15.5
	15	8.5	9.2	9.9	10.9	11.5	12.5	13.4	14.3	15.2	15.8	16.5	17.2
	3/16	9.5	10.3	10.8	11.6	12.9	14.0	14.9	15.9	16.9	17.6	18.4	19.1
	9	10.3	11.2	11.9	12.7	14.0	15.2	16.2	17.3	18.3	19.1	19.9	20.7
	13/64	11.0	12.0	12.6	13.5	14.9	16.2	17.3	18.5	19.5	20.3	21.2	22.0
	4	12.2	13.3	13.9	14.9	16.5	17.9	19.1	20.4	21.5	22.4	23.4	24.2
	7/32	14.2	15.4	16.4	17.4	19.1	20.6	22.1	23.4	24.7	25.8	26.8	27.8
	15/64	16.1	17.5	18.6	19.7	21.7	23.4	25.0	26.4	27.8	29.0	30.0	31.1
	1/4	18.1	19.6	20.6	22.0	24.1	26.0	27.7	29.2	30.7	32.0	33.2	34.2
	17/64	20.0	21.6	22.7	24.2	26.5	28.6	30.3	32.0	33.6	34.8	36.1	37.2
	9/32	23.2	25.2	26.5	28.1	30.6	32.8	34.8	36.6	38.2	39.6	40.8	42.0
	5/16	26.0	28.2	29.5	31.4	34.0	36.4	38.5	40.2	41.9	43.3	44.6	45.8
	11/32	28.2	30.5	31.9	33.8	36.6	39.0	41.0	43.2	44.8	46.0	47.2	48.5
	3/8	30.9	33.5	35.3	34.5	38.1	41.2	44.0	46.7	49.2	51.5	53.5	55.5
¾	5/16				37.8	41.7	45.0	48.0	51.0	53.6	56.0	58.2	60.1
	21/64	33.7	36.4	38.8	41.1	45.0	48.6	52.0	55.1	58.1	60.4	62.6	64.8
	11/32	38.8	42.0	44.3	47.2	51.8	55.8	59.3	62.6	65.8	68.6	70.9	73.3
	3/8	43.5	47.1	50.1	52.7	57.8	62.1	65.8	69.5	72.6	75.7	78.2	80.4
	13/32	47.6	51.4	54.6	57.5	62.4	67.2	71.3	75.0	78.4	81.4	83.6	86.3
	7/16	54.1	58.5	61.9	65.9	70.8	75.5	79.7	83.6	87.0	90.0	92.6	95.2
	1/2	58.8	63.5	67.9	67.4	74.0	80.4	85.8	90.8	95.8	100.0	104.0	107.8
1	57/64				71.8	79.0	85.2	91.1	96.6	101.8	106.0	110.2	114.1
	7/16	66.0	71.3	76.1	80.4	88.4	95.5	101.7	107.3	112.7	117.7	122.1	126.2
	15/32	72.7	78.5	83.6	88.5	97.0	104.1	111.0	117.5	123.0	128.2	133.0	137.2
	1/2	84.5	91.4	97.0	102.4	112.0	120.0	127.2	134.3	140.5	146.0	152.0	155.1
	9/16	93.8	101.5	108.0	113.9	124.0	132.0	140.0	147.0	153.5	159.0	164.2	168.4
	5/8	103.0	111.6	119.2	114.0	124.0	132.1	145.9	155.8	164.5	171.0	179.0	185.1
1¼	17/32				126.1	139.0	151.0	161.0	171.1	180.5	188.0	196.1	203.3
	9/16	112.7	122.0	130.9	137.5	152.8	164.1	175.0	185.5	196.0	204.0	212.4	220.6
	19/32	122.0	132.1	141.2	149.0	164.0	177.7	188.9	201.0	211.0	225.0		
	5/8	139.0	150.9	161.5	169.8	187.0	191.0	213.9	228.0				
	11/16	154.0	166.5	178.0	187.6	206.0	222.0						
	3/4	176.8	191.5	205.0	214.5								
	7/8	199.0	215.5	221.0									
1½	3/4												
	13/16	217.5											

PIPE SIZE	DRILL SIZE	SECTION 1 — CONNECT TO RISERS ACCORDING TO FIGS. 5A-5C.																											
		13-14	14-15	15-16	16-17	17-18	18-20	20-22	22-24	24-26	26-28	28-30	30-33	33-36	36-40	40-43	43-46	46-50	50-55	55-60									
½	7/64	3.4	3.6	3.7	3.9	4.0	4.2	4.5	4.7	4.9	5.2	5.4	5.6	5.9	6.3	6.5	6.7	7.1	7.4	7.8	7/64	½							
	33	3.7	3.9	4.1	4.2	4.4	4.6	4.9	5.1	5.4	5.6	5.8	6.1	6.4	6.8	7.1	7.4	7.7	8.1	8.4	33								
	32	4.2	4.4	4.5	4.7	4.9	5.0	5.5	5.8	6.0	6.3	6.5	6.9	7.2	7.6	7.9	8.2	8.6	9.0	9.4	32								
	31	4.8	5.0	5.2	5.4	5.5	5.9	6.2	6.6	6.9	7.2	7.4	7.8	8.2	8.7	9.0	9.4	9.8	10.3	10.8	31								
	1/8	5.2	5.4	5.6	5.8	6.0	6.4	6.8	7.1	7.5	7.8	8.1	8.5	8.9	9.5	9.8	10.2	10.7	11.2	11.7	1/8								
	30	6.2	6.4	6.7	6.9	7.2	7.6	8.1	8.5	8.9	9.3	9.6	10.1	10.6	11.2	11.7	12.1	12.7	13.3	13.9	30								
	29	6.8	7.1	7.4	7.6	7.9	8.4	8.9	9.3	9.8	10.2	10.6	11.2	11.7	12.4	13.0	13.3	14.0	14.7	15.3	29								
	9/64	7.7	8.1	8.4	8.7	9.0	9.6	10.1	10.6	11.2	11.6	12.1	12.7	13.3	14.1	14.7	15.2	15.9	16.7	17.5	9/64								
	26	8.5	8.9	9.2	9.6	9.9	10.5	11.1	11.7	12.3	12.8	13.3	14.0	14.7	15.6	16.1	16.7	17.5	18.4	19.2	26								
	24	9.2	9.6	10.0	10.3	10.7	11.4	12.0	12.6	13.2	13.7	14.3	15.1	15.8	16.8	17.4	18.1	18.9	19.8	20.8	24								
	5/32	10.0	10.4	10.8	11.2	11.6	12.3	13.0	13.7	14.4	15.0	15.5	16.4	17.2	18.2	18.9	19.6	20.5	21.5	22.5	5/32								
	20	10.8	11.2	11.7	12.2	12.6	13.4	14.2	14.9	15.6	16.2	16.9	17.8	18.7	19.8	20.5	21.3	22.3	23.4	24.5	20								
	19	11.9	12.4	12.9	13.4	13.8	14.7	15.5	16.3	17.1	17.8	18.5	19.5	20.4	21.7	22.5	23.4	24.4	25.6	26.8	19								
	11/64	13.4	13.9	14.5	15.0	15.6	16.5	17.5	18.4	19.2	20.0	20.6	21.9	23.0	24.4	25.3	26.2	27.4	28.8	30.2	11/64								
	15	14.8	15.4	16.1	16.6	17.3	18.4	19.4	20.2	21.2	22.2	23.1	24.4	25.5	27.1	28.1	29.1	30.5	32.0	33.5	15								
	3/16	16.5	17.2	17.9	18.6	19.3	20.5	21.6	22.7	23.8	24.8	25.7	27.1	28.4	30.2	31.3	32.4	34.0	35.7	37.4	3/16								
	9	18.0	18.8	19.6	20.3	21.0	22.3	23.6	24.8	25.9	27.1	28.1	29.6	31.1	32.9	34.2	35.4	37.1	39.0	40.8	9								
	13/64	19.2	20.0	20.8	21.6	22.4	23.8	25.3	26.4	27.7	28.8	29.9	31.6	33.1	35.0	36.4	37.8	39.5	41.5	43.4	13/64								
	4	21.3	22.2	23.1	23.9	24.8	26.4	27.8	29.3	30.6	31.9	33.1	34.9	36.6	38.8	40.3	41.8	43.8	46.0	48.2	4								
	7/32	24.7	25.8	26.8	27.9	28.8	30.6	32.3	34.0	35.6	37.1	38.5	40.5	42.4	44.9	46.7	48.6	50.8	53.4	55.8	7/32								
	15/64	28.0	29.2	30.4	31.6	32.6	34.7	36.7	38.5	40.4	42.1	43.7	46.1	48.3	51.2	53.2	55.2	57.7	60.6	63.4	15/64								
	1/4	31.2	32.6	34.0	35.3	36.4	38.8	41.0	43.1	45.0	47.0	48.8	51.5	54.0	57.2	59.4	61.6	64.5	67.7	70.8	1/4								
	17/64	34.5	35.9	37.4	38.8	40.2	42.7	45.0	47.4	49.5	51.7	53.6	56.6	59.3	62.8	65.4	67.7	71.0	74.9	77.8	17/64								
	9/32	40.0	41.7	43.4	45.1	46.6	49.6	52.4	55.1	58.1	60.2	62.4	65.8	69.0	73.2	75.9	78.8	82.4	86.6	90.6	9/32								
	5/16	44.6	46.5	48.5	50.3	52.0	55.3	58.4	61.4	65.9	67.0	70.0	73.4	77.0	81.5	84.0	87.8	91.3	96.5	101.0	5/16								
	11/32	48.3	50.3	52.4	54.3	56.2	59.8	63.2	66.4	69.5	72.5	75.3	79.4	83.2	88.2	91.5	95.0	99.3	104.3	108.2	11/32								
	3/8	53.6	56.0	58.2	60.5	62.5	66.5	70.2	73.8	77.4	80.6	83.6	88.2	92.5	98.1	102.0	105.7	110.5	116.2	121.5	3/8								
¾	5/16																				5/16	¾							
	21/64	58.4	60.9	63.5	65.8	68.0	72.4	76.4	80.4	84.1	87.6	91.0	95.8	100.4	106.5	110.7	114.7	120.0	126.1	132.0	21/64								
	11/32	67.2	70.0	73.0	75.8	78.3	83.2	88.0	92.5	96.9	101.0	104.9	110.6	116.0	122.8	127.3	132.2	138.4	145.3	152.2	11/32								
	3/8	75.2	78.4	81.6	84.8	87.6	93.2	98.5	103.5	108.2	113.0	117.2	123.8	129.1	137.3	142.6	148.0	154.8	162.6	170.2	3/8								
	13/32	82.1	85.5	89.2	92.5	95.6	101.8	107.5	113.0	118.1	123.2	128.0	135.0	141.3	150.0	155.7	161.4	169.0	177.5	185.8	13/32								
	7/16	93.0	96.9	101.0	104.7	108.3	115.2	121.8	128.0	134.0	139.7	145.0	153.0	160.0	170.0	176.2	183.0	191.4	201.0	210.5	7/16								
	1/2	101.9	106.1	110.5	114.7	118.7	126.3	133.2	140.1	146.5	153.0	158.8	167.4	175.3	186.0	193.0	200.5	209.6			1/2								
1	57/64																				57/64	1							
	7/16	114.2	119.0	124.0	128.7	133.0	141.5	149.6	157.1	164.5	171.2	178.0	188.0	196.8	208.5	216.5	224.5				7/16								
	15/32	125.7	131.0	136.4	141.5	146.4	155.8	164.5	173.0	181.0	188.8	196.0	206.8	217.0							15/32								
	1/2	145.9	152.0	158.2	164.1	169.9	180.6	190.8	200.6	210.0	218.5	226.3									1/2								
	9/16	161.8	168.5	175.5	181.9	188.2	200.1	211.3	222.3												9/16								
	5/8	179.9	187.3	195.0	202.1	209.0	222.5														5/8								
1¼	17/32																				17/32	1¼							
	9/16	196.3	204.5	213.0	221.3																9/16								
	19/32	212.5	221.8																		19/32								
	5/8																				5/8								
	11/16																				11/16								
	3/4																				3/4								
	7/8																				7/8								
1½	3/4																				3/4	1½							
	13/16																				13/16								

TABLE 3—CAPACITIES OF SMALL RISERS AND RISER BRANCHES. (PIPE SIZES FOR THE BRANCHES THE SAME AS FOR THE RISERS)

PIPE SIZES		CAPACITY IN EQUIVALENT SQUARE FEET OF DIRECT RADIATION				PIPE SIZES	
SUPPLY	RETURN	SECTION 4	SECTION 3	SECTION 2	SECTION 1	SUPPLY	RETURN
½	½	14.6	23.6	31.6	36.9	½	½
½	¾	19.9	31.9	42.5	49.6	½	¾
¾	¾	31.1	49.1	65.0	75.6	¾	¾
¾	1	40.7	64.2	85.0	99.0	¾	1
1	1	58.5	92.8	122.4	143.2	1	1
1	1¼	78.9	125.0	165.0	193.0	1	1¼
1¼	1¼	121.0	190.2	253.0	295.0	1¼	1¼
1¼	1½	145.2	237.0	304.0	354.0	1¼	1½
1½	1½	181.4	288.0	378.0	442.5	1½	1½

Section 1: the remaining 2/7 of the line lying between section 2 and the "boiler."

Step 6.

Select the proper sizes of "radiator" branches and orifices from Table 2. Include the expansion tank unless it is separately connected to the "boiler." The height above the mid-point of the "boiler" in feet is the vertical distance in feet from the average height of the "boiler" openings to the average height of the "radiator" openings. Thus, the mid-point of the "boiler" is usually considerably above the actual middle of the "boiler," and the measurement should be made either to the middle or the bottom of the "radiator," depending on whether the high or low supply connection is used. It will usually be found most convenient to begin with the "radiators" in section 4.

Step 7.

If the installation is more than one story high, select the proper sizes for the risers from Table 3 or Table 4 depending on the capacity required. Include the expansion tank unless it is separately connected to the "boiler."

Step 8.

Select the sizes for the mains by means of Table 5 using either the simple or the more accurate method. The *Distance from Mid-point of "Boiler" to Lowest Point of Supply Main* is the vertical distance in feet from the average height of the "boiler" openings to the middle of the supply main at the junction of this main with the "boiler" uptake. The return main may be either on the same level with the supply main or below it, *but the return main must never be above the supply main.*

TABLE 4—CAPACITIES OF LARGE RISERS AND RISER BRANCHES. (PIPE SIZES FOR THE BRANCHES LARGER THAN FOR THE RISERS)

PIPE SIZES				CAPACITY IN EQUIVALENT SQUARE FEET OF DIRECT RADIATION				PIPE SIZES			
SUPPLY RISER	RETURN RISER	SUPPLY RISER BRANCH	RETURN RISER BRANCH	SECTION 4	SECTION 3	SECTION 2	SECTION 1	SUPPLY RISER	RETURN RISER	SUPPLY RISER BRANCH	RETURN RISER BRANCH
1¼	2	2	2	232	369	483	567	1¼	2	2	2
2	2	2	2½	356	568	744	872	2	2	2	2½
2	2½	2½	2½	424	676	885	1040	2	2½	2½	2½
2½	2½	3	3	577	914	1197	1400	2½	2½	3	3
2½	3	3	3	705	1123			2½	3	3	3
3	3	3½	4	1020				3	3	3½	4
3	3½	4	4	1169				3½	3½	4	4

TABLE 5—CAPACITIES OF MAINS IN EQUIVALENT SQUARE FEET OF DIRECT RADIATION

DISTANCE FROM MID-POINT OF BOILER TO LOWEST POINT OF SUPPLY MAIN 3FT-4FT.

PIPE SIZES		LENGTH OF LINE IN FEET																					PIPE SIZES	
SUPPLY	RETURN	0-20	20-25	25-30	30-35	35-40	40-45	45-50	50-60	60-70	70-80	80-90	90-100	100-120	120-140	140-160	160-180	180-200	200-240	240-280	280-330		SUPPLY	RETURN
½"	½"	18	17	16	14	13	12	11	10	9	8	7	7	6	5								½"	½"
½"	¾"	24	23	21	19	18	17	16	14	13	11	10	10	8	7								½"	¾"
¾"	¾"	38	35	32	30	28	26	25	22	20	18	16	15	13	12	12	11	10	9	9	8		¾"	¾"
¾"	1"	49	45	42	39	36	34	32	28	26	23	22	20	18	17	15	14	13	12	11	10		¾"	1"
1"	1"	69	64	60	55	52	48	45	41	37	34	32	30	28	25	21	19	18	16	15	13		1"	1"
1"	1¼"	92	85	80	74	70	65	62	56	51	47	44	41	37	33	30	27	25	23	21	18		1"	1¼"
1¼"	1¼"	136	127	119	112	106	100	95	87	81	75	70	67	60	55	50	46	43	38	34	29		1¼"	1¼"
1¼"	1½"	162	151	141	133	125	119	114	104	97	90	85	81	73	67	61	58	54	48	42	36		1¼"	1½"
1½"	1½"	201	185	172	163	154	147	140	129	121	114	108	103	93	85	79	74	71	63	56	48		1½"	1½"
1½"	2"	262	246	230	215	202	192	184	166	157	147	140	133	122	112	108	100	95	84	75	64		1½"	2"
2"	2"	376	349	327	305	291	277	266	242	227	210	198	189	176	162	150	143	134	124	114	100		2"	2"
2"	2½"	460	423	398	365	349	332	319	296	279	261	250	236	219	201	188	175	165	148	137	120		2"	2½"
2½"	2½"	560	515	480	456	437	415	400	380	360	345	329	313	287	265	245	225	209	185	167	145		2½"	2½"
2½"	3"	700	650	611	580	557	531	510	483	455	433	412	391	357	332	311	292	278	241	218	197		2½"	3"
3"	3"	931	864	842	805	770	740	710	665	620	580	550	521	474	446	417	395	377	346	319	285		3"	3"
3"	3½"	1105	1046	994	946	910	874	840	787	737	693	659	628	575	538	504	479	457	417	383	342		3"	3½"
3½"	3½"	1359	1280	1215	1156	1111	1067	1030	965	912	865	825	788	731	683	645	610	580	526	480	428		3½"	3½"
3½"	4"	1521	1462	1392	1330	1274	1226	1177	1101	1040	989	947	912	853	803	761	723	686	619	559	488		3½"	4"
4"	4"	1805	1711	1635	1568	1495	1432	1375	1280	1210	1158	1115	1080	1025	977	931	889	845	756	673	570		4"	4"
4"	5"	2280	2170	2070	1990	1908	1821	1764	1630	1554	1481	1432	1387	1308	1242	1181	1115	1075	964	854	750		4"	5"
5"	5"	3090	2950	2830	2730	2620	2525	2445	2290	2170	2075	2000	1940	1808	1700	1605	1530	1460	1330	1210	1100		5"	5"
5"	6"	3720	3580	3450	3310	3200	3070	2980	2810	2680	2570	2480	2390	2250	2115	2005	1905	1820	1659	1526	1390		5"	6"
6"	6"	4660	4570	4460	4280	4110	3940	3800	3650	3510	3380	3265	3160	2970	2800	2660	2520	2420	2235	2080	1890		6"	6"

DISTANCE FROM MID-POINT OF HEATER TO LOWEST POINT OF SUPPLY MAIN 4 FT.-5 FT.

PIPE SIZES		LENGTH OF LINE IN FEET																					PIPE SIZES	
SUPPLY	RETURN	0-20	20-25	25-30	30-35	35-40	40-45	45-50	50-60	60-70	70-80	80-90	90-100	100-120	120-140	140-160	160-180	180-200	200-240	240-280	280-330		SUPPLY	RETURN
½"	½"	21	20	18	16	15	14	13	11	10	9	9	8	7	6								½"	½"
½"	¾"	28	26	24	22	21	20	19	17	15	14	12	11	9	8								½"	¾"
¾"	¾"	44	40	37	35	32	30	29	25	23	21	19	17	15	14	14	13	13	12	10	9		¾"	¾"
¾"	1"	57	52	49	45	42	39	37	33	30	27	25	23	21	19	17	16	15	14	13	12		¾"	1"
1"	1"	80	74	69	64	60	56	53	47	43	39	37	35	31	28	24	22	20	18	17	15		1"	1"
1"	1¼"	106	98	91	85	80	75	72	64	59	54	51	47	42	37	34	31	29	27	24	21		1"	1¼"
1¼"	1¼"	157	146	137	128	122	115	111	102	94	87	81	77	69	63	58	54	50	44	39	33		1¼"	1¼"
1¼"	1½"	187	174	163	154	144	137	132	120	112	104	98	93	83	76	71	66	62	55	49	42		1¼"	1½"
1½"	1½"	252	234	198	188	178	170	162	149	140	132	125	119	107	98	91	89	82	73	65	57		1½"	1½"
1½"	2"	303	280	261	245	233	222	212	195	181	175	162	153	141	129	120	113	110	97	87	74		1½"	2"
2"	2"	435	403	378	352	336	320	307	280	260	242	228	218	203	187	173	165	155	143	132	115		2"	2"
2"	2½"	524	480	450	421	403	384	368	342	322	301	285	272	250	230	215	202	188	171	158	139		2"	2½"
2½"	2½"	646	595	555	527	504	480	462	439	419	398	380	366	331	306	285	260	241	213	197	179		2½"	2½"
2½"	3"	808	750	705	670	643	614	590	558	526	500	476	452	417	384	358	332	310	278	257	238		2½"	3"
3"	3"	1075	1020	972	930	890	855	820	768	716	670	635	601	546	515	481	456	435	400	368	329		3"	3"
3"	3½"	1275	1210	1148	1090	1050	1009	970	908	850	800	760	725	664	621	582	554	527	481	443	395		3"	3½"
3½"	3½"	1568	1460	1403	1333	1282	1231	1190	1111	1050	998	955	910	845	788	745	705	670	607	555	494		3½"	3½"
3½"	4"	1760	1690	1610	1535	1470	1416	1360	1272	1200	1140	1093	1052	985	928	879	835	793	715	641	564		3½"	4"
4"	4"	2080	1978	1890	1810	1728	1668	1588	1480	1397	1337	1290	1250	1165	1130	1074	1025	975	873	776	659		4"	4"
4"	5"	2630	2500	2390	2300	2200	2110	2035	1880	1790	1710	1650	1600	1510	1433	1365	1288	1230	1111	986	866		4"	5"
5"	5"	3570	3400	3270	3150	3030	2920	2825	2640	2500	2395	2310	2225	2085	1965	1852	1760	1686	1535	1400	1270		5"	5"
5"	6"	4300	4140	4000	3840	3700	3570	3440	3240	3090	2870	2870	2760	2600	2440	2310	2200	2100	1914	1760	1605		5"	6"
6"	6"	5390	5280	5150	4950	4750	4550	4390	4210	4050	3900	3770	3650	3430	3235	3070	2910	2800	2580	2400	2180		6"	6"

Step 9.

If there is more than one elbow per main close to the "boiler," increase the size of this portion of the mains as follows:

a. For two elbows per main use one size larger than given in tables.

b. For three or four elbows per main use two sizes larger than given in the tables.

Step 10.

Select the sizes for the "boiler" uptakes and "boiler" returns from Table 6.

A Simple Method of Selecting Pipe Sizes for a One-Story Residence

Fig. 10 represents a layout that has been made in accordance with the foregoing general procedure, as follows:

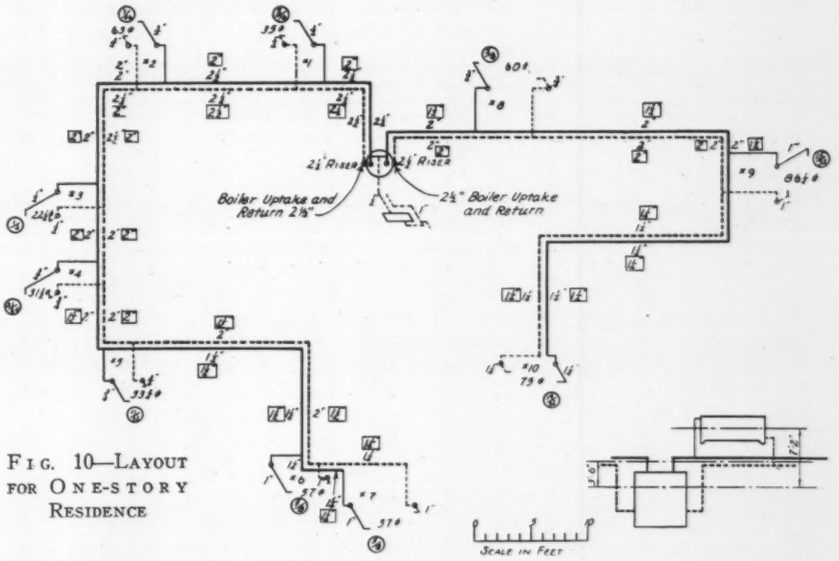


FIG. 10—LAYOUT FOR ONE-STORY RESIDENCE

Step 1.

It is assumed that the "radiators" have been selected and located as specified in step 1 of the *General Procedure in Designing a System.*

Step 2.

The piping is to correspond with the most common practice. Thus, the mains are to be side by side as in Fig. 7*A*. The "boiler" uptakes are to be separate as in Fig. 8*A*. If all the risers pass through reasonably large holes and are free to move in any horizontal direction, all the "radiators" may be connected according to Figs. 4*A* or 4*B*, but if No. 2, No. 5 or No. 9 has risers passing through snug holes or sleeves, each of these "radiators" having such relatively immobile risers should be connected according to Figs. 4*G* or 4*H*.

Step 3.

Checking the layout according to step 3, it will be found that the longer branch contains five elbows per main, but since the last of these occurs between the last two "radiators" on the line it may be neglected, which leaves four, the maximum number allowable. The right hand branch has four elbows per main which is also satisfactory.

Step 4.

Checking again according to step 4, it will be found that there is only one elbow per main close to the heater which, again, is as it should be.

Step 5.

The length of the left hand branch is 8 ft + 23 ft + 23½ ft + 18 ft + 13 ft + 7 ft or 92½ ft $\frac{92.5}{7} = 13.2$ ft. Therefore, section 4 consists of the 13.2 ft at the end farthest from the "boiler," section 3 consists of the next 26.4 ft., Section 2 consists of the next 26.4 ft., and section 1 consists of the remaining 26.4 ft. between section 2 and the "boiler."

Step 6.

According to step 5, "radiators" No. 7 and No. 6 lie in section 4 and since the average height of the inlet and outlet is 7 ft. 2 in. above the mid-point of the "boiler," the column headed 7-8 in Table 2, *Section* 4, must be used. Since the values given in this table are maximums, the first number larger than 57 must be selected. This number is 58.8 and by following the same line across the table to the left margin, it will be found that these "radiators" each require 1-in. "radiator" branches and a ⅝-in. orifice. In Fig. 10, the size of the drill required to bore an orifice is indicated by a number within a circle. "Radiator" No. 5 (33¾ sq ft) lies in section 3 and by using the column headed 7-8 in Table 2, *Section* 3, it will be evident that this "radiator" requires ¾-in. "radiator" branches and an 11/32-in. orifice. "Radiator" No. 4 lies almost exactly on the dividing line between section 3 and section 2. Since it is a small "radiator," it probably will be advisable to place it in section 3. This arrangement also distributes the "radiation" among the sections somewhat more uniformly than would be the case if it were placed in section 2. Accordingly, again using the column headed 7-8 in Table 2, *Section* 3, it will be found that 31½ sq ft require ¾-in. "radiator" branches and a 21/64-in. orifice. "Radiators" No. 3 and No. 2 lie in section 2, and by using the column headed 7-8 in *Section* 2 of Table 2, it will be found that No. 3 (22½ sq ft) requires ½-in. branches and a ¼-in. orifice, while No. 2 (63 sq ft) requires ¾-in. branches and a 7/16-in. orifice. "Radiator" No. 1 lies in section 1, and by using the column headed 7-8 in *Section* 1 of Table 2, it is apparent that a 35-sq ft "radiator" requires ½-in. branches and a 5/16-in. orifice. The foregoing data may be expressed in compact tabular form as follows:

"Radiator" Number	Size of "Radiator" in Equivalent Square Feet	Section of Line	Size of "Radiator" Branches	Size of Drill Required to Bore Orifices
7	57	4	1	5/16
6	57	4	1	5/16
5	33¾	3	¾	11/32
4	31½	3	¾	11/32
3	22½	2	½	¼
2	63	2	¾	7/16
1	35	1	½	5/16

The right hand branch is 66½ ft long. Section 4 is therefore 9.5 ft long and each of the other sections, 19.0 ft long. This places "radiator" No. 10 in section 4, "radiator" No. 9 in section 2, and "radiator" No. 8 in section 1. The required sizes for orifices and "radiator" branches may again be found from Table 2 and tabulated as follows:

"Radiator" Number	Size of "Radiator" in Equivalent Square Feet	Section of Line	Size of "Radiator" Branches	Size of Drill Required to Bore Orifices
10	75	4	1¼	11/32
9	86¼	2	1	11/32
8	60	1	¾	3/8

Step 7.

Does not apply to a one-story installation.

TABLE 6—CAPACITIES OF BOILER UPTAKES AND BOILER RETURNS

PIPE SIZES	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	7	8	PIPE SIZES
CAPACITY IN EQUIVALENT SQ. FT.	13	26	49	86	122	216	333	542	713	940	1520	2360	3220	4200	CAPACITY IN EQUIVALENT SQ.FT.

TABLE 7—EQUIVALENT LENGTH FACTORS

USE WITH 3FT-4FT. TABLE								USE WITH 4FT-5FT. TABLE							
3'-0"	3'-2"	3'-4"	3'-6"	3'-8"	3'-10"	4'-0"	ACTUAL DISTANCE ABOVE MIDPOINT	4'-0"	4'-2"	4'-4"	4'-6"	4'-8"	4'-10"	5'-0"	5'-2"	5'-4"	5'-6"
1.000	0.943	0.891	0.845	0.802	0.765	0.730	FACTOR	1.000	0.953	0.915	0.878	0.843	0.812	0.782	0.754	0.730	0.707

Step 8.

The simple method of selecting sizes for mains is to apply the data in Table 5, and for each part of the line to select the sizes that have a capacity equal to or greater than the load which that particular part of the line is required to carry. Since the lowest portion of the mains is 3½ ft above the mid-point of the "boiler," the required data are to be found in the upper half of Table 5. The left hand branch is 92½ ft long, so the column headed 90-100 is used. Thus, the portion of the left hand branch lying between "radiators" No. 7 and No. 6 must carry 57 sq ft. The first number in the 90-100 column larger than 57 is 67. Following the line under 67 to either the right or the left margin, it will be found that the supply main must be 1¼ in. and the return main must also be 1¼ in. Proceeding in a similar manner, the following sizes for all portions of the mains are obtained:

"RADIATORS" BETWEEN WHICH MAINS LIE	REQUIRED CAPACITY	SUPPLY SIZE	RETURN SIZE
No. 7 and No. 6	57	1¼	1¼
No. 6 and No. 5	114	1½	2
No. 5 and No. 4	147¾	2	2
No. 4 and No. 3	179¼	2	2
No. 3 and No. 2	201¾	2	2½
No. 2 and No. 1	264¾	2½	2½
No. 1 and Heater	299¾	2½	2½
No. 10 and No. 9	75	1¼	1¼
No. 9 and No. 8	161¼	2	2
No. 8 and Boiler	221¼	2	2

Step 9.

Since there is only one elbow per main close to the "boiler" step 9 does not apply.

Step 10.

The sizes for the "boiler" uptakes and "boiler" returns are obtained from Table 6. Since the left branch carries 299¾ sq ft of "radiation," the required size is 2½-in. The right branch carries 221¼ sq ft and therefore the required size for it is also 2½-in.

In most cases it will be found convenient to combine all of the foregoing steps in one table as follows:

"RADIATOR" NO.	SIZE OF "RADIATOR" IN SQ FT	SECTION OF LINE	SIZE OF "RADIATOR" BRANCHES	SIZE OF DRILL REQUIRED TO BORE ORIFICES	REQUIRED CAPACITY FOR MAINS	SIZE OF SUPPLY MAINS	SIZE OF RETURN MAINS	SIZE OF "BOILER" UPTAKES AND BOILER RETURNS
7	57	4	1	⅝	57	1¼	1¼	
6	57	4	1	⅝	114	1½	2	
5	33¾	3	¾	11	147¾	2	2	
4	31½	3	¾	11	179¼	2	2	
3	22½	2	½	¼	201¾	2	2½	
2	63	2	¾	7/16	264¾	2½	2½	
1	35	1	½	5/16	299¾	2½	2½	
								2½
10	75	4	1¼	11	75	1¼	1¼	
9	86¼	2	1	11	161¼	2	2	
8	60	1	¾	5/16	221¼	2	2	
								2½

A More Accurate and Economical Method

In making an accurate design, steps 1 to 7 are the same as for the simple design but in step 8, by using a more elaborate method of selecting the sizes of the mains, a considerable saving can be made. It will at once be recognized that since the pipe sizes recommended in the 3 Ft—4 Ft part of Table 5 are satisfactory when the lowest portion of the mains is 3 ft above the mid-point of the heater, they must be larger than necessary when the distance above the mid-point is more than 3 ft. In order to aid in the selection of these smaller sizes Table 7 has been provided.

Example. Suppose that a line is actually 100 ft long but that the distance from the mid-point of the heater to the lowest portion of the mains is 3 ft 4 in. instead of an even 3 ft. On the 3 Ft-4 Ft side of Table 7 under the heading 3 *ft*-4 *in.* will be found the equivalent length factor 0.891. Consequently, the equivalent length of this 100-ft line is 0.891 × 100 ft or 89.1 ft and the sizes may be taken from the 80-90 column of Table 5 instead of from the 90-100 column of that table and a saving usually will result.

If a still further reduction in pipe sizes is desired, it can be obtained provided the designer is sufficiently careful. The capacities given in Table 5 are the maximums that can be carried without exceeding the available friction per foot, but whenever any part of the line is carrying less than its capacity load as determined by Table 5, its friction is less than the maximum allowable which means that it will be permissible for some other part of the mains to carry more than its rated capacity, provided the extra friction caused by this excess is equal to or less than the friction saved where the load is less than the rated capacity. Since the friction in the mains varies approximately as the square of the load carried, it is very important that this part of the work be done carefully. The friction increase due to a given overload is much greater than the friction decrease due to an equal underload.

The general method for designing mains accurately may be divided into 4 steps as follows:

Step 1.

Find the equivalent length of the line by means of Table 7.

Step 2.

Make a tentative selection of pipe sizes from Table 5 using the equivalent length of the line and selecting the pipe sizes having the capacity nearest to the required capacity. When in doubt as to which of two sizes is nearer, choose the larger.

Step 3.

Calculate the friction factors for the various parts of the mains, making a separate calculation for each part that carries a different load or has different pipe sizes from the adjacent parts. The friction factor is calculated by means of the following formula:

$$F = L - \frac{LS^2}{C^2}$$

where

F = the friction factor.

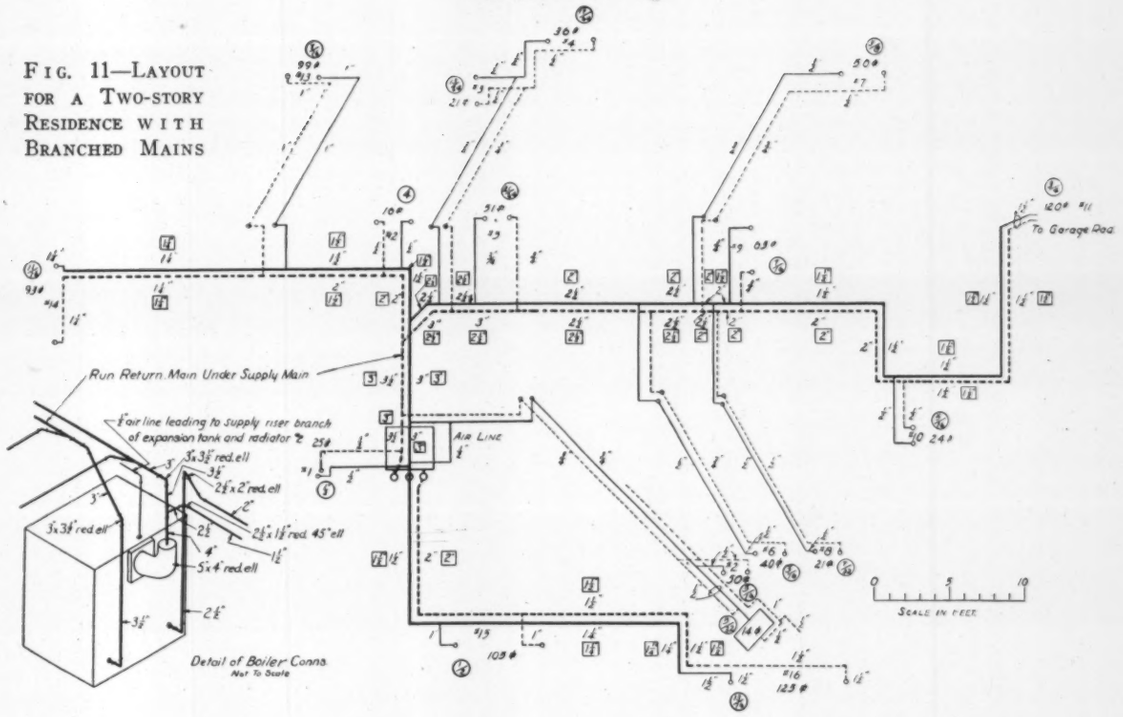

Run Return Main Under Supply Main
Air Line leading to supply riser branch of expansion tank and radiator #2
Air Line
To Garage Rad.
Detail of Boiler Conns.
Not To Scale
Scale in Feet

FIG. 11—LAYOUT FOR A TWO-STORY RESIDENCE WITH BRANCHED MAINS

L = the length, including elbow equivalents, of the part of the mains being checked. If the supply and return mains are not the same length, or do not contain the same number of elbows, take the average in each case. The equivalent of an elbow in feet of pipe may be taken as twice its nominal diameter in inches.

C = the capacity of the mains as given in Table 5.

S = the actual load the mains must carry in equivalent square feet.

Step 4.

Arrange the positive factors in one column and the negative factors in another. If the sum of the positive factors is equal to or greater than the sum of the negative factors, the design is satisfactory. If the sum of the negative factors is larger, some of the pipe sizes must be increased. Often it is possible to effect a considerable saving by being careful in selecting the places where the mains are to be overloaded.

In applying these methods to the left branch of the one-story layout of Fig. 10, the steps are as follows:

Step 1.

Since the lowest portion of the mains is 3 ft-6 in. above the mid-point of the heater, the equivalent length factor is found in Table 7 under the heading 3 *ft*—6 *in.* This factor is 0.845 and since the actual length of the line was previously found to be 92½ ft, the equivalent length is 0.845×92.5 or 78.2 ft. Therefore, the pipe sizes are to be taken from the 70-80 column of the 3 *Ft*—4 *Ft.* part of Table 5.

Steps 2, 3 *and* 4.

These three steps are best arranged in a tabular form similar to that to the left, below:

Several points regarding the foregoing tabulation should be noted. The *first* is that in selecting the pipe sizes given in columns 3 and 4 the mains nearest the heater were overloaded, that is, the actual load on this part of the mains is greater than their rated capacities as given in Table 5. This can nearly always be done and usually will save money because it reduces the quantity of large pipe needed.

The *second* relates to column 5. Since the equivalent of an elbow in feet of pipe is twice its nominal diameter in inches, the average equivalent for two mains, each of which contains an elbow, will be simply the sum of the nominal diameters of the two elbows. The numbers in the column in question were obtained by simply adding to the average length of the mains the nominal diameter of each elbow. Thus, the first three numbers in this column were obtained as follows:

$$7 + 1\tfrac{1}{4} + 1\tfrac{1}{4} = 9\tfrac{1}{2}$$
$$28\tfrac{1}{2} + 1\tfrac{1}{2} + 1\tfrac{1}{2} = 31\tfrac{1}{2}$$
$$9 + 1\tfrac{1}{2} + 2 = 12\tfrac{1}{2}$$

The *third* point to note is that the calculations have been so arranged that the numbers in each column can be obtained from the numbers in other columns either by a single slide rule operation or by a single subtraction.

The tabulated calculations for the more accurate design of the right branch of Fig. 10 are as follows:

1	2	3	4	5	6	7	8	9
"Radiators" Between Which Mains Lie	Load (S)	Pipe Sizes		Length of Mains (L)	Rated Capacity (C)	$\frac{LS^2}{C^2}$	Friction Factors $L - \frac{LS^2}{C^2}$	
		Supply	Return				Positive	Negative
7-6	57	1¼	1½	9½	75	5.48	4.02	.....
6-5	114	1½	1½	31½	114	.....	0	0
5-4	147¾	1½	2	12½	147	12.63	.....	0.13
4-3	179¼	2	2	7¼	210	5.28	1.97	.....
3-2	201¾	2	2	18½	210	17.05	1.45	.....
2-1	264¾	2	2½	14¼	261	14.63	.....	0.38
1-Boiler	299¾	2	2½	16½	261	21.75	.....	5.25
						Totals	7.44	5.76

$66.5 \times 0.845 = 56.2$ Equivalent Length of Branch

"Radiators" Between Which Mains Lie	Load	Pipe Sizes		Length of Mains (L)	Rated Capacity (C)	$\frac{LS^2}{C^2}$	Friction Factors $L - \frac{LS^2}{C^2}$	
		Supply	Return				Positive	Negative
10-9	75	1¼	1¼	31½	87	23.40	8.10	.....
9-8	161¼	1½	2	22½	166	21.25	1.25	.....
8-Boiler	221¼	1½	2	12½	166	21.75	.....	9.250
							9.35	9.250

Steps 9 and 10 are the same as for the simple method.

The pipe sizes for the mains that were obtained by the more accurate method are shown in squares on Fig. 10.

Branched Line in a Two-Story Residence

In Fig. 11 is shown the layout of a two-story plant that has a branched line. In making the layout it has been assumed that the return main is to be run directly under the supply main, since this usually is the simplest and cheapest method of handling a branch. The "radiators" are to have the high supply connection of Figs. 4*B* and 5*B*. The riser branches are to be according to Fig. 6*A* and the boiler is to be connected according to Fig. 8*C*.

In designing a system of this type it is necessary to begin with the longer branch. This branch is divided into sections just as though the shorter branch did not exist. Thus, since it is 65 ft from the top of the "boiler" uptake to the connections of radiator No. 11, the lengths of the sections are as follows:

Section 4: $\frac{65}{7} = 9.3$ ft

Sections 3, 2 and 1: $65 \times 2/7 = 18.6$ ft

To obtain the lengths of the sections in the shorter branch, measure the lengths of both branches beginning at the junction and multiply the lengths of the sections in the longer branch by the ratio of the length of the shorter branch to the length of the longer branch. Thus, the distance along the mains from the junction to "radiator" No. 14 is 26 ft and the distance from the junction to "radiator" No. 11 is 55 ft. Consequently, the lengths of the sections in the shorter branch are:

Section 4: $9.3 \times 26/55 = 4.4$ ft

Sections 3, 2 and 1: $18.6 \times 26/55 = 8.8$ ft

Total length (for purpose of selecting sizes of mains): $65 \times 26/55 = 30.7$ ft.

The short line at the bottom of the layout is designed just the same as for a one-story system.

In tabulating the calculations for a two-story installation, an extra column giving the height of the "radiator" above the mid-point of the "boiler" is needed and also columns for riser sizes which are sometimes larger than the "radiator" branches of second floor "radiators." The tabulated calculations are shown below.

The method of obtaining the numbers in the first column of the tabulation below, particularly the 30.7, has already been explained. The numbers in the second column correspond to the "radiator" numbers given on the layout. The numbers in the third column give the section in which each "radiator" lies and the fourth column gives the vertical distances from the mid-point of the "boiler" to the "radiator." Since the high supply connection is used this distance is measured to the middle of the "radiators." There is a slight variation because some of the "radiators" are taller than others.

The fifth column headed *Square Feet for "Radiator" Branches* contains simply the rating of each "radiator" in equivalent square feet. The sixth column headed *Square Feet for Risers and Riser Branches* gives the load that must be carried by each riser. In case the riser carries more than one "radiator," the ratings must

Vertical Distance from Mid-Point of Boiler to Lowest Point in Supply Main 3½ Ft

Length of Main Branch In Feet	Radiator Number	Section of Line	Height Above Mid-Point of "Boiler"	Square Feet for "Radiator" Branches	Square Feet for Risers and Riser Branches	Square Feet for Mains	Table 2		Table 3		Table 5 3 Ft.-4 Ft.		Table 6	
							Drill Size for Orifice	"Radiator" Branches	Risers and Riser Branches		Mains		Boiler Uptake	Boiler Return
									Supply	Return	Supply	Return		
65	11	4	8′ 0″	120	.....	120	3/4	1½	.....	.....	1½	1½	.....	.....
	10	3	7′ 4″	24	.....	144	5/8	½	.....	.....	1½	2	.....	.....
	9	2	8′ 0″	65	.....	209	7/8	¾	.....	.....	2	2	.....	.....
	8	2	17′ 0″	21	21	230	#33	½	½	½	2	2½	.....	.....
	7	2	17′ 0″	50	50	280	5/8	½	¾	¾	2½	2½	.....	.....
	6	2	17′ 0″	40	40	320	5/8	½	½	¾	2½	2½	.....	.....
	5	1	7′ 6½″	51	.....	371	#11	¾	.....	.....	2½	3	.....	.....
	4	1	16′ 6½″	36	.....	.....	#17	½	.....	.....	.....	.....	.....	.....
	3	1	17′ 0″	21	57	428	#11	½	¾	¾	2½	3	.....	.....
30.7	14	4	7′ 6″	93	.....	93	9/16	1¼	.....	.....	1¼	1¼	.....	.....
	13	2	17′ 0″	99	99	192	7/16	1	1	1	1½	2	.....	.....
	12	2	7′ 6″	16	.....	208	4	½	.....	.....	1½	2	.....	.....
			Junction of Branches											
	.....	.....	.....	.....	.....	636	.....	.....	.....	.....	3	3½	.....	.....
65	Exp.	1	26′ 0″	14	14	.....	#33	½	½	½	.....	.....	.....	.....
	2	1	17′ 0″	50	64	700	5/8	½	¾	¾	3	3½	.....	.....
	1	1	8′ 4½″	25	.....	725	#11	½	.....	.....	3	3½	.....	4
30	16	4	7′ 8½″	125	.....	125	9/16	1½	.....	.....	1¼	1½	.....	.....
	15	2	7′ 6½″	105	.....	230	½	1	.....	.....	1½	2	.....	2½
	.....	.....	.....	.....	.....	955	.....	.....	.....	.....	.....	.....	5	.....

be added as in the case of "radiators" No. 3 and No. 4.

The seventh column headed *Square Feet for Mains* contains the load that each part of the mains must carry. These loads are obtained by adding successively the loads contributed by each pair of riser or "radiator" branches where they enter the mains. The remaining columns containing orifice and pipe sizes are obtained by means of the tables. At the head of each column is indicated the table which applies to that particular column.

It will be noted that the supply connection to "radiator" No. 16 is larger than the main that it joins. This is due to the shortness of the line and does no harm, but it could have been avoided as will be explained in the answer to question 19.

Economical Method Applied to a Branch Line

Since it is usually most economical to overload a system close to the "boiler," it will be advisable in the case of a branched line to calculate the part between the junction and the "boiler" first, choosing the pipe sizes so as to make the total friction factor negative and then to so choose the pipe sizes for each branch that each shall have a total positive friction factor at least as large as the total negative friction factor of the part between the junction and the "boiler." The lowest part of the supply main is 3½ ft above the mid-point of the heater so the equivalent length factor according to Table 7 is 0.845.

The tabulated calculations are as follows:

$65 \times 0.845 = 54.9$, equivalent length of 65-ft line.
$30.7 \times 0.845 = 25.9$, equivalent length of 30.7-ft line.
$30 \times 0.845 = 25.4$, equivalent length of 30-ft line.

"Radiators" Between Which Mains Lie	Load (S)	Pipe Sizes		Length of Mains (L)	Rated Capacity (C)	$\frac{LS^2}{C^2}\frac{(LSS)}{(CC)}$	Friction Factors $L-\frac{LS^2}{C^2}$	
		Supply	Return				Positive	Negative
Junc.-2	636	3	3	6.0	665	5.48	0.52	.....
2-1	686	3	3	2.5	665	2.66	.....	0.16
1-Boiler	711	3	3	3.5	665	4.00	.....	0.50
							0.52	0.66
						Total		0.14
11-10	120	1½	1½	20.0	129	17.32	2.68	.....
10-9	144	1½	2	18.0	166	13.56	4.44	.....
9-8	209	1½	2	1.5	166	2.38	.....	0.88
8-7	230	2	2	1.5	242	1.36	0.14	.....
7-6	280	2	2½	3.5	296	4.09	.....	0.59
6-5	320	2	2½	10.0	296	11.70	.....	1.70
5-4 & 3	371	2½	2½	3.5	380	3.34	0.16	.....
4&3-Junc.	428	2½	2½	5.5	380	6.99	.....	1.49
							7.42	4.76
14-13	93	1¼	1¼	14.0	119	8.56	5.44	
13-12	192	1½	1½	8.0	172	9.97	.....	1.97
12-Junc.	208	1½	2	7.5	230	9.30	.....	1.86
							5.44	3.38

The calculations for the foregoing tabulation are so similar to the accurate main calculations for the one-story installation that they can probably be readily understood without comments. No attempt was made to make a more accurate design for the short line at the bottom of the figure because the design was evidently already as close as it was practical to endeavor to attain.

Economical Method Applied to a Branch of Three-Story Installation

The calculations for a three-story installation will be illustrated by means of the experimental plant shown in Fig. 1. In this case, since the lowest point on the supply main is exactly 4 ft above the mid-point of the "boiler," there is no need to use Table 7. Also, for the same reason the lower half of Table 5 is used instead of the upper half. The friction factor calculations have been combined with the other calculations, all of which are given in Table 8.

Methods Used in Calculating

In making Table 5 giving the capacities of mains, the pressure head available at a point 3 ft above the mid-point of the heater was first calculated. Ten per cent of this was set apart to overcome the friction in the "boiler" fittings, including the uptake and return, and the elbows that connect them to the mains. Fifteen per cent more was set apart to be available at the end of the line to make certain that the last "radiator" started in the right direction. The remaining 75 per cent was reserved for the mains themselves. Allowance was made for four elbows in each main. The "boiler" temperature differential was taken as 40 deg in order to obtain the available pressure head, which was 174×3 or 522 mil-inches. To get the friction in the mains, the quantity of water flowing through them was taken as the quantity that would be required to flow through the "radiators" at an average temperature differential of 28.5 deg when they were dissipating heat at their full rated capacity.

After the capacities of the mains having the same size supply and return were calculated, the capacities of the mixed sizes were calculated by the formula:

$$C = \frac{1.905\ C_1\ C_2}{C_1 - C_2}$$

where

C = the capacity of the mixed size.
C_1 = the capacity of a line having both pipes the same size as the supply main of the mixed line.
C_2 = the capacity of a line having both pipes the same size as the return main of the mixed line.

When the capacity of one of the sizes is double that of the other, formula (1) is accurate to about 0.2 per cent and it is sufficiently accurate for practical purposes over a considerably wider range than any for which it is likely to be used.

In order to obtain the pressure heads available for the various "radiator" and riser branches the average pressure between the mains at a height of 4 ft above the heater was computed for each section of the line. The pressure heads for the first floor "radiator" branches were then computed by adding the head generated in the "radiator" circuit to the head present in that particular part of the mains. The "radiators" in the first section of a line were to run at a temperature differential of 37.5 deg, those in the second section at 34.5 deg, those in the third section at 29.5 deg, and those in the fourth section at 22.5 deg.

After the available pressure heads had been deter-

TABLE 8—CALCULATIONS FOR A THREE-STORY INSTALLATION

Radiator Number	(L) Distances Between Branches Including Elbow Equivalents	Section of Line	Height Above Mid Point of Boiler	Square Feet for Radiator Branches	Square Feet for Risers and Riser Branches	(S) Square Feet for Mains	Table 2		Table 3		Table 5 (3 Ft.-4 Ft.) (90-100) (Column) Mains		Table 6		(C) Rated Capacity of Mains	$\frac{LS^2}{C^2}$ $\frac{(LSS)}{(CC)}$	$\frac{L-LS^2}{C^2}$ Friction Factors	
							Drill Size for Orifice	"Radiator" Branches	Risers and Riser Branches				Boiler Uptake	Boiler Return			Plus	Minus
									Supply	Return	Supply	Return						
35	.....	4	25′ 5″	15	15	.....	7/32	½	½	½	.....	.....	.....	.....	.....	.....	.....	.....
25	.....	4	16′ 5″	13¾	28¾	.....	11/64	½	¾	¾	.....	.....	.....	.....	.....	.....	.....	.....
15	.....	4	7′ 6″	13¾	.....	.....	7/32	½	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	13	.....	.....	.....	.....	42½	.....	.....	.....	.....	1	1¼	.....	.....	47	10.63	2.37	.....
34	.....	4	25′ 5″	15	15	.....	7/32	½	½	½	.....	.....	.....	.....	.....	.....	.....	.....
24	.....	4	16′ 5″	74¼	89¼	.....	9/64	1	1¼	1¼	.....	.....	.....	.....	.....	.....	.....	.....
14	.....	4	7′ 6″	74¼	.....	.....	11/64	1¼	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	24	.....	.....	.....	.....	206	.....	.....	.....	.....	2	2	.....	.....	218	21.45	2.55	.....
33	.....	3	25′ 5″	75	75	.....	11/64	¾	1	1	.....	.....	.....	.....	.....	.....	.....	.....
23	.....	3	16′ 5″	74¼	149¼	.....	½	¾	1¼	1¼	.....	.....	.....	.....	.....	.....	.....	.....
13	.....	3	7′ 6″	13¾	.....	.....	7/32	½	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	31	.....	.....	.....	.....	369	.....	.....	.....	.....	2½	2½	.....	.....	361	32.40	.....	1.40
32	.....	2	25′ 5″	55	55	.....	9/64	½	¾	¾	.....	.....	.....	.....	.....	.....	.....	.....
22	.....	2	16′ 5″	38½	93½	.....	7/32	½	1	1	.....	.....	.....	.....	.....	.....	.....	.....
12	.....	2	7′ 6″	38½	.....	.....	9/64	¾	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	26	.....	.....	.....	.....	501	.....	.....	.....	.....	3	3	.....	.....	601	18.05	7.95	.....
31	.....	1	25′ 5″	75	75	.....	3/16	½	¾	¾	.....	.....	.....	.....	.....	.....	.....	.....
21	.....	1	16′ 5″	74¼	149¼	.....	11/64	¾	1	1¼	.....	.....	.....	.....	.....	.....	.....	.....
11	.....	1	7′ 6″	74¼	.....	.....	9/64	¾	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	14	.....	.....	.....	.....	724½	.....	.....	.....	.....	3	3	.....	.....	601	20.30	.....	6.30
Heater	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	4	4	.....	Totals	12.87	7.70

mined, the capacities of the orifices and "radiator" branches for the first floor "radiators" were determined by the formula:

$$S = T\sqrt{\frac{PH}{1 - K\ P}} \qquad (2)$$

where

S = the capacity in equivalent square feet

T = the temperature differential in degrees Fahrenheit

H = the available pressure head in milinches of water

K = the friction head in milinches of water that would exist in the pipes and fittings, exclusive of the orifice, if they were connected to one equivalent square foot of heating surface operating at a temperature differential of one degree.

$$P = \frac{0.334\ (D - 0.046)^4\ d^4}{d^4 - (D - 0.046)^4} \qquad (3)$$

where

D = the diameter of the orifice in inches

d = the actual internal diameter of the pipe in inches

From a hydrodynamic standpoint, formula (3) is not accurate because it is designed not simply to give the quantity of water which must flow through the "radiator" per second in order to produce a given friction head, but to admit a larger quantity of water to a small "radiator" or one high above the "boiler" and thus to compensate for the greater cooling in pipes that are unusually small or long. From the data given in the first part of this paper, it can be seen that it accomplished this purpose very well in the case of the experimental plant.

The risers in the fourth section have a maximum allowable friction of 16 milinches of water per vertical foot or 8 milinches per foot in each pipe when both pipes are the same size. In the third section the riser friction is 22½ milinches per vertical foot. In the second section it is 28 milinches per vertical foot, and in the first section it is 32 milinches per vertical foot. As in the case of the mains, the capacities for pipes of equal size were computed first and the mixed sizes calculated by means of the formula:

$$\frac{1.905\ C_1\ C_2}{C_1 - C_2}$$

The friction in the riser branches is held below an allowable maximum of 280 milinches in section 4, the sizes of one or both of the branches being made larger than the sizes of the corresponding risers whenever necessary. In the other sections the allowable maximum friction is greater but the general method is the same. The pressure head across an upper floor "radiator" branch cannot be determined as accurately as the pressure head across a first floor "radiator" branch because of the variable character of the riser friction. Mean values were therefore taken and the calculations made by means of formula (2).

Also, since one of the primary objects of these tables is to obtain costs as low as is consistent with reliability, a great deal of attention had to be given to safety factors.

Questions and Answers

Question 1: Why are ceiling "radiators" recommended for use in basements whenever basement "radiators" are required?

Answer: Principally because ceiling "radiators" which lie entirely above the mains can be listed as an integral part of Table 2, while some other type of "radiator" would require special tables.

Question 2: Is it not a disadvantage to have tables that require so many different sizes of drills?

Answer: It is, but one means of reducing the safety factors required is to figure orifice capacities in small steps. At any

rate it probably will never be necessary to buy very many drills at one time because the average installation requires only from six to twelve different sizes.

Question 3: Is it worth while to go to all the trouble of making the more accurate design for the mains?

Answer: It is always worth while to find the equivalent length of a line. Whether it is worth while to figure the friction factors is questionable. It may be that for small buildings it will be more satisfactory to be careful to err on the side of safety when selecting the sizes for the mains. The capacities of the mains may be exceeded in places but these excesses must be small, much smaller than the underloads that are intended to off-set them. Also, the length of the line affected should be less for the overloaded than for the underloaded portions. However, the friction factor method should not be discarded without giving it an extended trial. It is much more exact than pure guessing and when a slide rule is used, the procedure is comparatively simple.

Question 4: How often may trouble be expected with systems designed according to these tables?

Answer: There is every reason to expect that circulation troubles will be almost unknown except in cases where a serious mistake has been made. However, no practical tests have been made on systems where some of the "radiators" are more than 30 ft above the mid-point of the "boiler" and it is always possible that an untried portion of a table may not work out exactly as expected.

Question 5: What should be done in case trouble does develop?

Answer: It is usually necessary to place smaller orifices in the "radiators" that are functioning satisfactorily. Sometimes covering the supply piping will improve conditions, but covering the return mains or branches is likely to make matters worse. Sometimes it is advisable to take the orifice out of the "radiator" that is not functioning properly but the installation of smaller orifices in the hot "radiators" is the most reliable remedy.

Question 6: In case a carefully designed system fails to circulate properly, what is the most probable cause?

Answer: Probably some of the orifices have been omitted. The omission of some of the orifices is almost certain to cause trouble and the omission of only one may sometimes result in serious consequences.

Question 7: Will the "radiators" on the end of a line get as hot as those near the "boiler"?

Answer: In a fairly tall building it is expected that the "radiators" in section 4 will be somewhat hotter than those in sections 1 and 2, while in low rambling buildings the "radiators" in the sections close to the "boiler" are likely to become somewhat hotter than those in section 4.

Question 8: Why was section 4 made shorter than the other sections?

Answer: Because, in an average system, there is a tendency for heating surface to be *bunched* at the end of a line. Since section 4 "radiators" require more water than similar "radiators" in other sections, too much heating surface in section 4 might overload the mains and cause trouble. This is especially true where the mains branch.

Question 9: Can these tables be extended so as to apply to taller buildings?

Answer: It seems probable that they can. So far as the writer has been able to ascertain, the only limitation upon the height to which hot water can be successfully used for heating is the limitation due to the hydrostatic pressure caused by a high column of water, provided proper tables have been developed.

Question 10: Can these tables be used to design systems having welded mains?

Answer: These tables can be used to design systems having welded mains provided the elbows in the mains are of the long sweep type and the riser branches and "radiator" branches are screwed. If welded riser and "radiator" branches are used there is some likelihood that it may be necessary to reduce orifice sizes in order to compensate for the fact that the long sweep welded elbows have lower friction than the screwed elbows.

Question 11. In reference to covered mains, would it seriously affect the circulation or the balance of a system if the supply pipes only were covered for one-story jobs and supply piping and return risers only for two- or three-story jobs?

Answer: On a line where most of the "radiators" are small, covering the supply piping would be likely to improve the circulation. On a multiple-story line having mostly large "radiators," it would probably cause the fourth section "radiators" to have a slightly higher average temperature than the remainder of the system. The average temperature of the system would, however, be higher and the temperature differentials lower than would be the case were no covering used. To sum up, covering all the supply piping and return risers, if any, would probably improve some systems and would never have any noticeable adverse effect provided the return mains and riser branches were left bare.

Question 12: If it is necessary to locate a small "radiator" on the end of a line, is there anything that can be done in designing the system to insure satisfactory operation?

Answer: There are certain precautions that can be taken in advance and an easy remedy that can be applied later if necessary. They are as follows:

1. Be sure that the mains, especially the return main, are large enough throughout the entire line.

2. In the doubtful "radiator," use an orifice two or three sizes larger than called for by the tables.

3. If necessary, cover all the supply piping leading to the "radiator" including both branches and risers, if any, and also cover the portions of the supply main where the pipe is small.

Question 13: Why does a closed system under pressure operate at a higher temperature differential than an open system?

Answer: If the open and closed systems were operated at the same temperatures there would be no appreciable difference in their temperature differentials, but when the closed system is raised to a temperature higher than is ever used for an open system, it dissipates heat more rapidly than does the open system and consequently its temperature differentials are higher.

Question 14: In the case of the tests on the experimental plant, why were the hottest and coolest "radiators" different when the system was closed than when it was open?

Answer: This was probably due to the fact that the "radiators" stood on a scaffold in one large room, with the top floor "radiators" very close to the ceiling. Under these conditions there was a tendency for the air surrounding the top floor "radiators" to be too hot, which, in turn, caused the top floor "radiators" to come to a higher temperature than they otherwise would have. This tendency was much more pronounced with the closed system because the high temperature "radiators" heated the air to a higher temperature before it rose to the ceiling. Also, when the system was operated at high temperatures there was a tendency for the basement to overheat. The combined effect of these two tendencies was to raise the average temperatures of the "radiators" on the third floor and in the fourth section of the line and, as a result, the hottest and coolest "radiators" were not the same ones that had been hottest and coolest when the system was operated at lower temperatures. In addition to this, the steam pressure on the converter could not be held constant and consequently the temperature of the water leaving the converter was somewhat variable. As a result, a difference of two or three degrees between the temperatures of any of the "radiators" had very little if any significance.

Question 15: Referring to Fig. 4, why may "radiators" connected according to *A, D, G,* and J, be expected to give somewhat better circulation than "radiators" connected by the other methods illustrated?

Answer: When a "radiator" is connected according to *C, F, I,* or *L,* there is a tendency for eddies to be set up within the "radiator" so that the water goes up through some of the tubes

and down through others. As a result, the incoming hot water is mixed with partly cooled water and a lower average "radiator" temperature results. This effect is most pronounced in the "radiators" close to the "boiler" where the temperature differences are large. When a "radiator" is connected according to *B, E, H,* or *K,* the velocity head of the entering water is frequently sufficient to carry the water entirely across the top of the "radiator" and part way down the return end before its energy is exhausted. As a result, the bottom corner of the "radiator" on the supply end is slow to heat. This condition is most noticeable in the fourth section and upper floor radiators where the circulation is rapid. In the connections shown at *A, D, G,* and *J,* the velocity head of the entering water is mostly used to overcome the greater friction encountered by the water that takes the longest possible path through the "radiator" so that the "radiator" temperature is likely to be more uniform than in either of the other cases. Also, when the high supply connection is used there is likely to be a smaller difference in temperature between the hottest "radiator" and the coolest because the heads generated in the "radiator" circuits are larger which tends to overcome the unavoidable differences in pressure that exist between the mains in different parts of the same section.

Question 16: Why should the measurement of the vertical distance above the mid-point of the heater be made to the middle of the radiator when the high supply connection is used and to the height of the unions when the low supply connection is used?

Answer: Because when the high supply connection is used the water in the "radiator" being cooler (on the average) and therefore heavier than the water in the top of the supply "radiator" branch creates a pressure head which should be taken into account when the size of the orifice is selected. When the low supply connection is used, the water in the "radiator" presses equally on both the supply and return "radiator" branches and consequently the water in the "radiator" cannot add anything to the available pressure head no matter how tall the "radiator" may be.

Question 17: Did the experimental plant give more air-binding trouble when a high pressure was used?

Answer: Yes. There was no trouble at all when the expansion tank was open but when it was closed "radiator" No. 34 became air-bound. A close examination of this radiator revealed a small previously unnoticed leak. Apparently when the system was cold and under a subatmospheric pressure, this leak had admitted the air which blocked the circulation to this "radiator" the next time the system was put in operation. It should be stated, however, that the experimental plant was never operated continuously for more than a few hours at a time and for that reason, no very reliable conclusions as to air-binding can be drawn at present.

Question 18: In step 5 under *General Procedure in Designing a System* is it necessary to reduce elbows to equivalent feet of pipe before proceeding to divide the mains into sections?

Answer: No, it is not necessary to take account of elbows in sectioning the mains, but if a branch occurs close to the dividing line between two sections the locations of the elbows in the mains may help the designer to decide in which section that particular branch had better be placed.

Question 19: In the case of "radiator" No. 16 in Fig. 11, the supply "radiator" branch is larger than the supply main which feeds it. Could the size of this branch be safely reduced to the size of the main?

Answer: Yes. If the orifice size is increased to ⅞-in., the circulation will probably be just as good as if the tables had been followed exactly. The ⅞-in. orifice with the 1¼-in. supply branch and the 1½-in. return branch will give almost exactly the same friction as a 13/16-in. orifice with both branches 1½-in.

Question 20: If the size of the "boiler" uptake or "boiler" return called for by the tables is larger than the corresponding "boiler" opening, what should be done?

Answer: This is not likely to happen, but if it should, the small "boiler" connections can be compensated for by using larger mains than called for by the tables. There are so many fittings in the "boiler" connections that the mains will need to be enlarged for a considerable distance in order to fully compensate for an uptake or "boiler" return that is too small.

Question 21: Should the expansion pipe and the circulating pipe always be ½-in. and the overflow always 1-in., as shown in Fig. 9*A* and Fig. 9*B*?

Answer: Yes. According to Table 2, a pair of ½-in. risers can take care of 36.9 equivalent square feet. Since there are usually fewer elbows in the connections to an expansion tank than in the connections to a "radiator" and also since it is always desirable that the expansion tank dissipate as little heat as possible, these sizes should be ample.

Question 22: Is it necessary to put a check valve in the vent pipe when the expansion tank connections are made exactly according to Fig. 9*A* or Fig. 9*B*?

Answer: If either Fig. 9*A* or Fig. 9*B* is followed exactly, experience with the experimental plant indicates that a check valve will be unnecessary. When the cold water feed was wide open and the system full of water the reducing tee at the top of the overflow acted like an aspirator and air was drawn down through the vent pipe. Upon another occasion when the average temperature of the system was about 220 F, the valve between the expansion tank and the vent pipe was opened wide, but nothing came out of the vent except an almost imperceptible amount of flash steam.

In case it should be necessary to run the upper portion of the overflow horizontally for a considerable distance, a check valve might be required.

Question 23: In case the expansion tank is connected to risers and requires an orifice, how should the size of the orifice be determined?

Answer: There is considerable latitude in determining the size of an expansion tank orifice and it need not be done accurately. A simple and satisfactory method is to select the orifice that will take care of a "radiator" the equivalent feet of which are equal to the actual external area of the tank.

Conclusions

The experimental results reported in this paper appear to be very satisfactory and there is sufficient reason to believe that every radiator on an open system designed according to the tables given can be counted on to deliver 150 Btu per hour per equivalent square foot without approaching the boiling point. Also, it seems safe to assume that closed systems designed with only 10 per cent more heating surface than is required for steam will function satisfactorily provided the safety valve is set to blow off when the pressure at the top of the highest radiator is 15 or 20 lb per square inch, gage.

It is not known whether these pipe sizes are the most economical that can be incorporated in practical tables, but it is believed that they are fairly close to that ideal. In considering the economy of pipe sizes it should be remembered that one of the prime requisites of an economical installation is that there shall be no expensive changes or adjustments to make after the system is once installed.

In conclusion, acknowledgment is made to H. M. Hart and R. W. Shields for their help in putting this paper into the form that would be most useful to the industry, and also to Professors Crawford and Brewer of the Mechanical Engineering Department of Agricultural and Mechanical College of Texas for their kindness in donating the use of their laboratory and tools.

Steam Condensation an Inverse Index of Heating Effect

By A. P. Kratz[1] (*MEMBER*) and M. K. Fahnestock[2] (*NON-MEMBER*)

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Research Laboratory.

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. This work, which has been in progress for the past five years, is a part, and a continuation of a general research program devoted to the study of Heating Rooms by Direct Steam and Hot Water Radiators, and carried on in the Department of Mechanical Engineering under the general supervision of A. C. Willard, professor of heating and ventilation and head of the department. Acknowledgment is due to E. L. Broderick, research assistant in mechanical engineering, for valuable assistance in the collection and preparation of the data.

THE object of this part of the investigation was to make a comparison of the performance of several types of direct cast-iron radiators using steam as the heating medium, when the temperature of the air in the test room was maintained at approximately 70 F at the breathing level, with the performance of the same types of radiators when the temperature was maintained at 68 F at a level 30 in. from the floor. It was suggested that such a comparison might justify recommendations for the adoption of a standard for testing such radiators.

Description of Apparatus

All tests were conducted in the low temperature testing plant which has been completely described in previous publications.[3] In brief, this plant, as shown in Figs. 1 and 2, consists of two rooms erected within a larger room having 6-in. cork walls, floor and ceiling. The latter room is equipped with direct expansion refrigerating coils, and each of the two test rooms presents two walls of standard frame construction exposed to any desired temperature maintained in the cold room. One wall of each test room contains a pair of double hung windows and the other wall a standard door with a glass panel. In addition, arrangements are made so that the floors and ceilings of the test rooms may be exposed to any desired temperatures. Air movement over the walls is obtained by means of fans.

The plant is thoroughly equipped with thermocouples and temperature recorders in order to permit all observations being made without the necessity for entering the rooms and thus disturbing conditions at any time during the period of a test. The radiators are connected with a single pipe for steam and return condensation, and the condensation is weighed as shown in Fig. 2. A steam separator is installed in the line to each radiator, and an electrical superheater further insures the use of dry steam. Provision is also made for venting the radiators to prevent the accumulation of air.

The temperatures at the various levels within the test rooms are observed by means of thermocouples made from No. 22 B and S gage copper and constantan wire and supported on a standard in the center of the room as shown in Fig. 2. A study of temperature conditions within the test rooms proved that the center reading at a given level is representative of the average temperature at that level.

The cast-iron radiators used for the present series of tests consisted of (1) a 6-section, 26-in., 5-tube radiator, as shown in Fig. 3; (2) a 10-section, 26-in., 3-tube radiator, as shown in Fig. 4; (3) a 13-section, 17-in., single tube panel radiator of the type shown in Fig. 5; and (4) a 3-section wall type of radiator as shown in Fig. 6, each section having a nominal rating of 7 sq ft. More complete dimensions and details for setting are given in the inset in Fig. 8. When the temperature at the 30-in. level was maintained at 68 F as compared with 70 F maintained at the 5 ft level, it was necessary to increase radiator (1) to 9 sections, radiator (2) to 12 sections, and radiator (3) to 16 sections, but no increase in radiator (4) was found necessary.

Test Procedure

In all cases the temperature in the cold room was maintained at about -2.5 F and one of the exposed walls of the test room was subjected to an equivalent wind velocity of approximately 10 mph. The temperature of the air above the ceiling of the test room was maintained at 62 F and the temperature of the air 3 in. below the floor about 2 deg higher than that 3 in. above the floor. No test observations were made until conditions had

[1] Research Professor in Mechanical Engineering, University of Illinois.

[2] Special Research Associate in Mechanical Engineering, University of Illinois.

[3] University of Illinois Engineering Experiment Station Bulletins Nos. 192 and 223. Also Transactions of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Vol. 35, 1929, pages 79-89.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

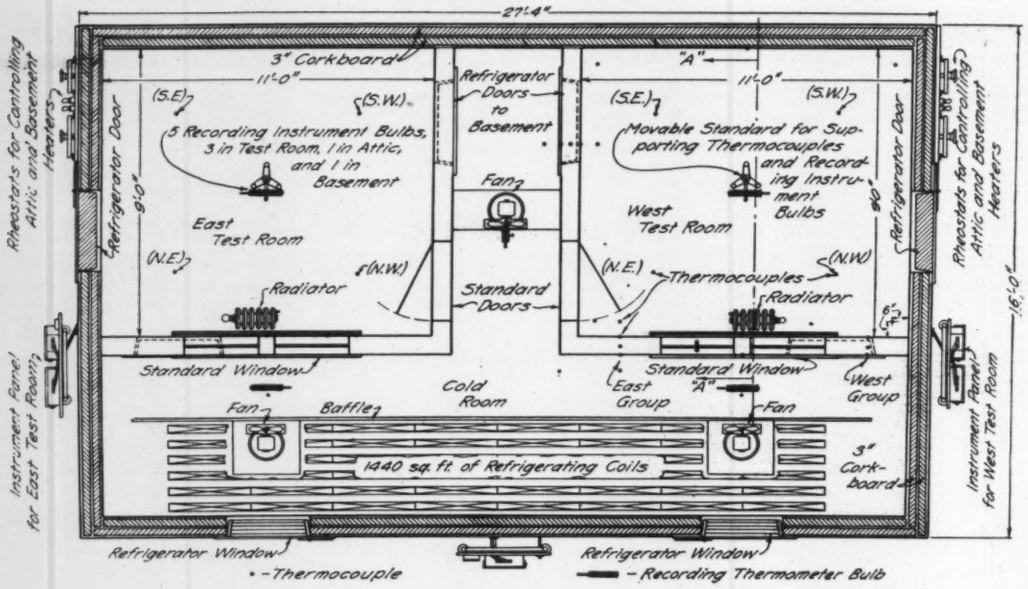


FIG. 1—PLAN SECTION OF LOW TEMPERATURE TESTING PLANT

remained constant for several hours, as indicated by the readings of the thermocouples on the inside wall surfaces. This required about 20 hours after a change in radiators was made. When the required thermal constancy had been attained, the condensate was weighed over a period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-minute increments of weight. At the end of each test a separate test was run to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

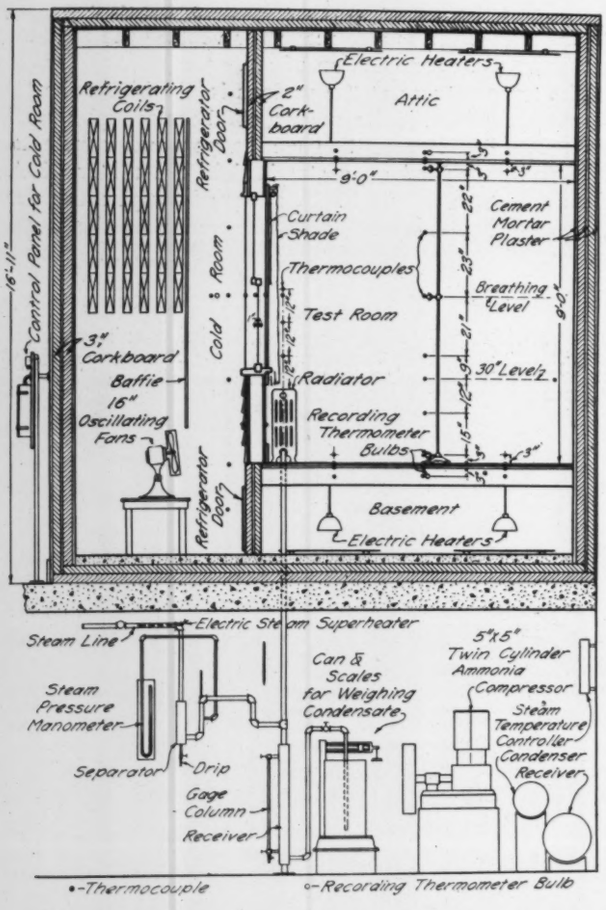


FIG. 2—ELEVATION SECTION OF LOW TEMPERATURE TESTING PLANT

Results of Tests

The results of the tests for which the temperature of the air at the breathing level (5 ft above the floor) was maintained at approximately 69 F are shown in Fig. 7. From these curves it is evident that while the radiators exhibited wide differences in characteristics relative to temperature distribution within the test room, the steam condensation required to maintain 69 F at the breathing level was practically the same for all of the radiators.

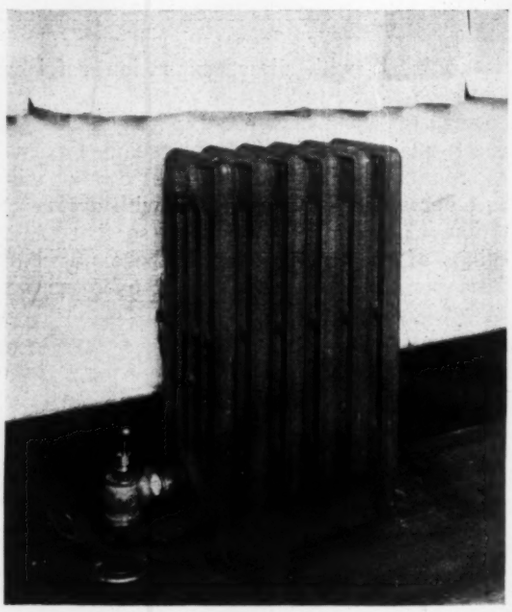


FIG. 3—6-SECTION, 26-IN., 5-TUBE RADIATOR

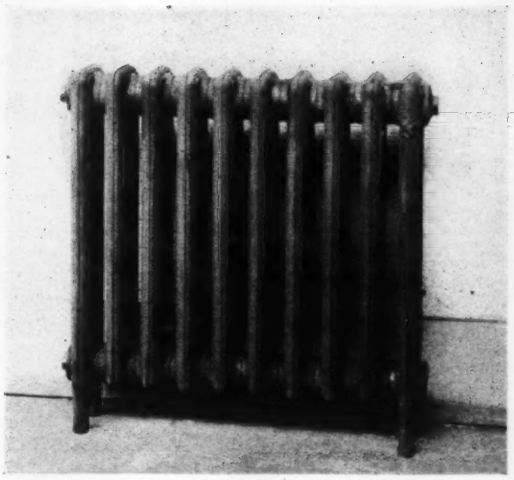


FIG. 4—10-SECTION, 26-IN., 3-TUBE RADIATOR

The maximum deviation in condensation was only 3.5 per cent.

Wide deviations in the degree of comfort produced in the room by the different radiators were apparent. With the exception of the wall type radiator, when the radiators were operated to maintain 69 F at the breathing level, the room was decidedly too cool for comfort. Some discomfort undoubtedly resulted from the fact that the inside surface temperature of the exposed walls was approximately 55 F, but in the case of the panel and the two tubular radiators, by far the greatest amount of discomfort was caused by the low temperatures in the zones below the breathing level, particularly in the zone from the 30-in. level down to the floor.

The differences between the steam condensations were not great enough to justify an attempt to evaluate the practical *heating effect* of the different radiators in terms of the steam condensation alone. The greatest temperature difference from floor to ceiling and the lowest temperatures between the floor and the 30-in. level were obtained with the 5-tube radiator. In this case the temperature at the floor was only 53.8 F and at the 30-in. level was only 59.9 F. The room was most uncomfortable when this radiator was used. The temperature at the breathing level obtained with this radiator was approximately one degree lower than the 69 F obtained with the other radiators. A slight increase in the size, sufficient to bring the breathing level temperature to 69 F would have increased the condensation slightly and increased the temperatures at all levels in the room by approximately one degree, but would not have appreciably increased the comfort. Thus, while the positions of the curves representing the temperatures at the levels below the breathing level afford a means of ranking the different radiators in the order of comfort produced, or *heating effect*, they constitute an intangible and rather unsatisfactory measure inasmuch as the statement of the results is entirely qualitative and not quantitative.

The 3-section wall type of radiator was the only one that produced a satisfactory degree of comfort in the room with the breathing level at 69 F. The reason for this, as previously mentioned, was that the temperature in all zones below the breathing level were higher than those for the other radiators. It was particularly noted (see Fig. 7) that the temperature at the 30-in. level was approximately 68 F. This indicated a possibility that if the sizes of the other types of radiators used were increased by amounts necessary to raise the temperature at the 30-in. level to 68 F in every case, that each radiator would maintain practically the same degree of comfort in the room, and a tangible measure of the heating effect might be obtained from the increase in condensation resulting from the increase in size necessary to obtain the 68 F at the 30-in. level.

The results of the tests with the air at the 30-in. level maintained at approximately 68 F are shown in Fig. 8. In all cases the room was comfortable and it is at once evident that the magnitudes of the steam condensations arrange themselves in the same order that radiators would be ranked from an inspection of the room temperature gradient curves. The total spread between steam condensations was increased to 21.5 per cent as compared with the 3.5 per cent obtained with the air at the breathing level maintained at 69 F. The 5-tube radiator, with the greatest temperature difference between the floor and

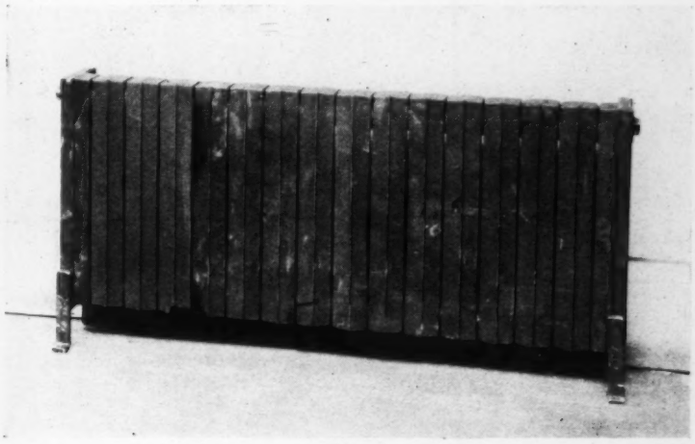


FIG. 5—16-SECTION, 17-IN., 1-TUBE PANEL RADIATOR

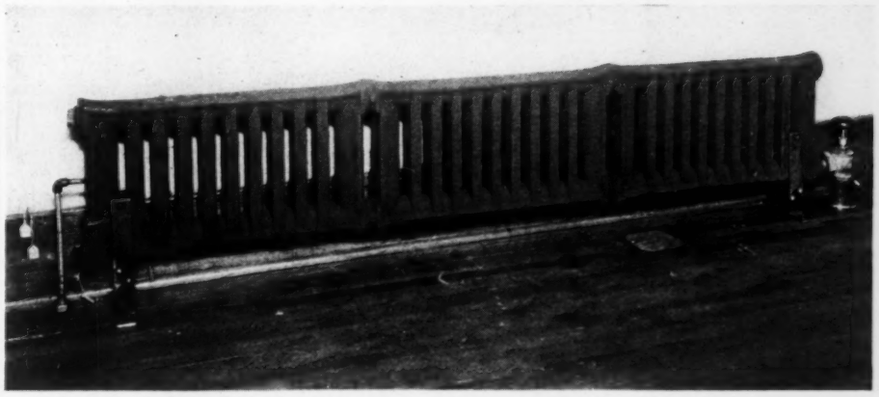


FIG. 6—3-SECTION WALL RADIATOR

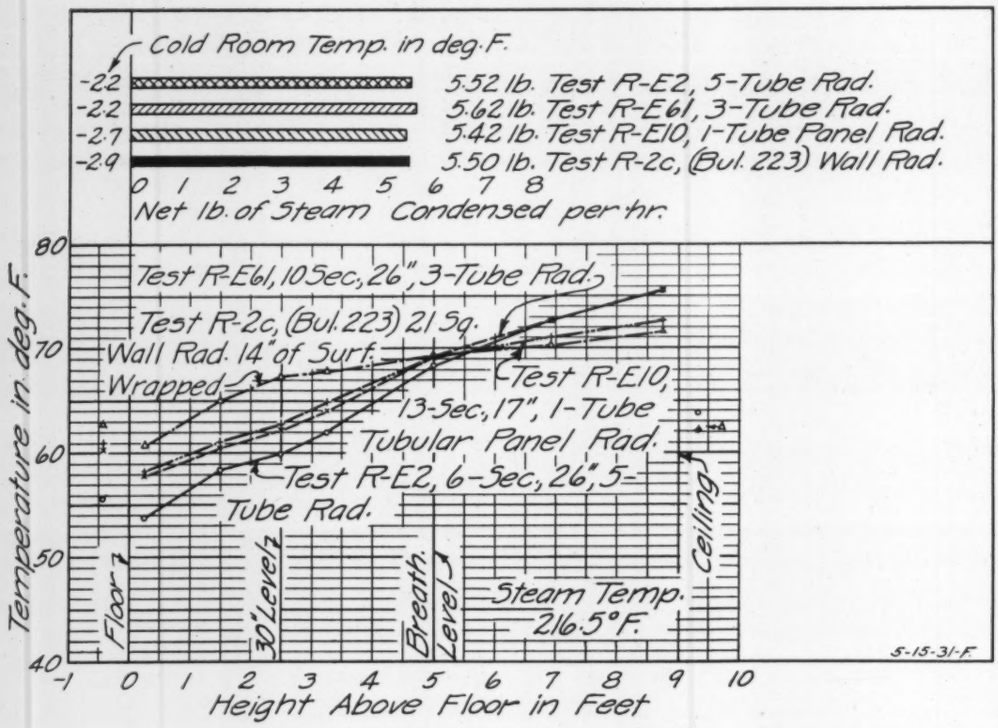

Cold Room Temp. in deg. F.
-2.2
-2.2
-2.7
-2.9
5.52 lb. Test R-E2, 5-Tube Rad.
5.62 lb. Test R-E61, 3-Tube Rad.
5.42 lb. Test R-E10, 1-Tube Panel Rad.
5.50 lb. Test R-2c, (Bul. 223) Wall Rad.
0 1 2 3 4 5 6 7 8
Net lb. of Steam Condensed per hr.
Test R-E61, 10 Sec., 26", 3-Tube Rad.
Test R-2c, (Bul. 223) 21 Sq. Wall Rad. 14" of Surf. Wrapped
Test R-E10, 13-Sec., 17", 1-Tube Tubular Panel Rad.
Test R-E2, 6-Sec., 26", 5-Tube Rad.
Floor
30" Level
Breath. Level
Ceiling
Steam Temp. 216.5° F.
5-15-31-F.
Temperature in deg. F.
80
70
60
50
40
-1 0 1 2 3 4 5 6 7 8 9 10
Height Above Floor in Feet

FIG. 7—ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF C. I. RADIATORS WITH A COMMON 60-IN. LEVEL TEMPERATURE

the ceiling, also gave the greatest steam condensation, while the 3-section wall type, with the least temperature difference, gave the least condensation. For the other radiators, the condensation was also in the order of the temperature difference. From Table 1 it is also evident that a consistent reduction in condensation occurred with a reduction in difference in temperature between the floor and ceiling.

The temperature at the 30-in. level obtained with the wall type radiator was approximately one degree Fahrenheit lower than that for the other radiators. A slight increase in the size of the radiator, sufficient to raise the temperature at the 30-in. level one degree Fahrenheit would have increased the temperature at all levels by the same amount, thus giving a new curve parallel to the old one and one degree higher. Hence, the tempera-

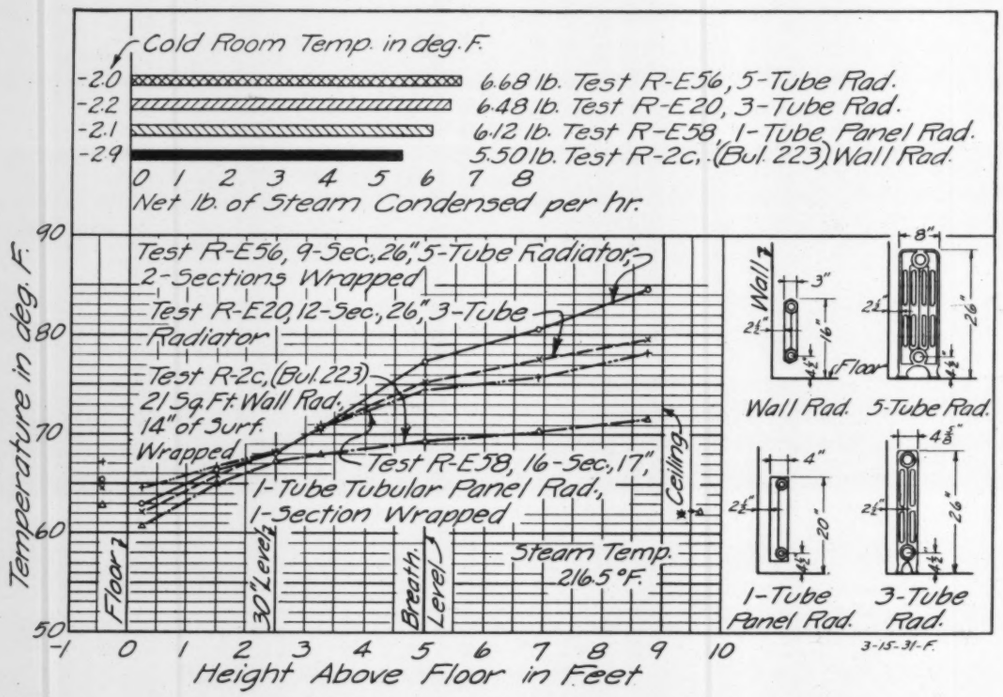


FIG. 8—ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF C. I. RADIATORS WITH A COMMON 30-IN. LEVEL TEMPERATURE

TABLE 1—COMPARISON OF STEAM CONDENSATIONS AND TEMPERATURE DIFFERENCES BETWEEN FLOOR AND CEILING FOR A CONSTANT AIR TEMPERATURE OF 68 F AT THE 30-IN. LEVEL

Type of Radiator	Temperature Difference Ceil.-Floor (F)	Steam Condensation (Lb Per Hr)	Ratio of Each Value in Columns (2) and (3) to Preceding Value	
			Temperature Difference	Steam Condensation
(1)	(2)	(3)	(4)	(5)
5-tube.....	21.6	6.68	.....	.....
3-tube.....	17.5	6.48	0.810	0.970
1-tube panel..	13.8	6.12	0.788	0.943
3-section wall.	10.8	5.50	0.782	0.900

ture difference between the floor and ceiling would remain the same and the ratio of temperature differences shown in the last figure of column 4 in Table 1 would not change. The steam condensation, however, would increase slightly over the 5.50 shown in column 3, thus increasing the ratio of condensations shown as the last figure in column 5 and making it more nearly in conformity with the next to the last figure, or 0.943. Therefore, if exact agreement had been obtained in the temperature at the 30-in. level for all of the radiators even greater quantitative consistency than that actually shown in Table 1 would have resulted.

Conclusions

From the results of these tests, the following conclusions may be drawn:

1 It is apparent that maintaining a temperature of 68 F at the 30-in. level is a more acceptable standard for stating conditions under which direct radiators shall be tested in the type of plant used than maintaining a temperature of 70 F at the breathing level.

2 With 68 F at the 30-in. level, the steam condensation obtained in the test plant used seems to be a measure of the effectiveness of the direct radiators tested, in providing for human comfort, or, more briefly, condensation becomes an inverse index of *heating effect*.

Classification of Coals

Up to the present time no one has laid down any definite system for classifying coal on the basis of relative percentages of anthraxylon and attritus as determined by actual measurement with the aid of the microscope. Such a system of classification is desirable in carbonization research because of the difference in coking properties of anthraxylon and attritus. It has been decided by the U. S. Bureau of Mines, therefore, to make arbitrary divisions of classification as follows:

Coals having a ratio of anthraxylon to attritus of more than 3:1 are called anthraxylous; those with a ratio between 3:1 and 1:1 are called anthraxylous-attrital; those with a ratio between 1:1 to 1:3 are called attrital-anthraxylous and coals in which the ratio of anthraxylon and attritus is less than 1:3 are called attrital. In other words, an anthraxylous coal consists of more than 75 per cent anthraxylon, and an attrital coal is more than 75 per cent attritus. It is believed that this system of classification will be of some use in the comprehensive research which the Bureau of Mines is now conducting on the carbonizing properties of American coals in cooperation with the American Gas Association.

Engineering Societies Monographs

The Engineering Societies Library has completed arrangements for the publication of a series of technical treatises to be known as *Engineering Societies Monographs*. The editorial supervision of this series will be in the hands of a committee consisting of H. W. Craver, Director of Engineering Societies Library, *Chairman*, and two representatives appointed by each of the following Engineering Societies: *American Society of Civil Engineers, American Institute of Mining and Metallurgical Engineers, American Society of Mechanical Engineers* and *American Institute of Electrical Engineers*.

For many years those who have been interested in the publication of papers, articles and books devoted to engineering topics have been impressed with the number of important technical manuscripts which have proved too extensive, on the one hand, for publication in the periodicals or proceedings of engineering societies, or in other journals, and of too specialized a character, on the other hand, to justify ordinary commercial publication in book form.

No adequate funds or other means of publication have been provided in the engineering field for making these works available to the profession. In other branches of science, certain outlets for comparable treatises have been available, and besides, the presses of several universities have been able to care for a considerable number of scholarly publications in the various branches of pure and applied science.

Experience has, however, demonstrated the value of proper introduction and sponsorship for such books. Without assuming responsibility for the authors' statements, the engineering societies, through suitable representatives, who have examined a manuscript, will furnish this introduction. In so doing, they perform a service to their members and, incidentally, to other persons seeking credible information.

The Engineering Societies Library will share in any profits made from publishing the Monographs; but the main interest of the Societies is service to their members and the public. With this aid the publisher is willing to adventure the production and distribution of selected books that would otherwise be commercially impracticable. He has prospects of losses on some and modest profits on others.

Engineering Societies Monographs will not be a series in the common use of that term. Physically they will have similarity, but there will be no regular interval in publication nor uniformity in price, nor relation in subjects. What books are printed and when, will by the nature of the enterprise depend upon what manuscripts are offered, estimation of their usefulness, cost of production in book form, and possibilities of distribution. The aim is to make promptly accessible to many users information which otherwise would be long delayed in reaching more than a few, in the wide domains of engineering.

Second International Heating and Ventilating Exposition at Cleveland

IT IS evident that the Second International Heating and Ventilating Exposition, to be held in Cleveland, January 25, 29, 1932, at the same time as the Annual Meetings of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers*, is well on its way to duplicate and even surpass the success of the first show at Philadelphia in 1930.

So many developments in heating, ventilating, and air conditioning practice and equipment have been reported since the previous exposition that the displays at the Cleveland Show are being awaited with keen anticipation, since many of them undoubtedly will be in the nature of premier presentations of the latest apparatus and devices.

A new Advisory Committee for the Exposition has been announced by the Society, consisting of:

Walter Klie, *Chairman*
H. P. Gant, *Vice-Chairman*

D. S. Boyden	J. F. McIntire
A. C. Edgar	J. C. Miles
C. Gottwald	H. C. Murphy
C. V. Haynes	M. F. Rather
C. H. B. Hotchkiss	E. A. Scott
W. C. Kammerer	W. E. Stark
F. P. Keeney	F. R. Still
F. A. Kitchen	T. A. Weager

A. W. Williams

The exposition will be managed by Charles F. Roth of the International Exposition Co.

The forthcoming show will have a central location in the Cleveland Public Auditorium Annex right in the heart of the Cleveland business district and within a few blocks of the leading hotels. An outstanding feature of the exhibits, particularly those having coal, gas and oil-burning equipment, will be the actual operation of the various devices. It will be recalled that this feature distinguished the previous exposition in Philadelphia and aroused wide comment.

An idea of the comprehensiveness of the Second International Heating and Ventilating Exposition is to be seen in the fact that it will embrace separate sections, one for the heating and ventilating industry in general, an oil burner section, a gas section, and a warm air furnace section.

In addition to the endorsement of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the Exposition will have the active support of the *American Oil Burner Association* and the *American Society of Refrigerating Engineers*. All three of these organizations will have booths at the Show, making the Exposition in fact representative of every branch of heating and ventilating.

The fact that the *American Society of Refrigerating Engineers* will hold its annual meeting in Cleveland simultaneously with that of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has made it possible to arrange for a joint session of both societies.

Among the organizations and companies which have already contracted for nearly one half of the available space at the Second International Heating and Ventilating Exposition are:

Aerofin Corp.
Aerologist, The
Air Reduction Sales Co.
American Air Filter Co.
American Gas Association
American Oil Burner Assn.
American Society of Heating and Ventilating Engineers
American Society of Refrigerating Engineers
American Temperature Indicating Co.
Ames Iron Works
Ames Pump Co., Inc.
Armstrong Machine Works
Armstrong Cork Co.
A. G. A. Testing Laboratory

B-Line Boiler Co.
Barber-Colman Co.
Barber Gas Burner Co.
Barnes & Jones
Bishop & Babcock Sales Co.
Brownell Co., The
Bryant Heater & Mfg. Co.
Buckeye Blower Co.

Carrier Corp.
Carrier Engineering Corp.
Carrier-Lyle Corp.
Century Electric Co.
Century Engineering Corp.
Chicago Pump Co.
Cleveland Gas Burner & Appliance Co.
Cleveland Steel Products Corp.
Cook Electric Co.
Coppus Engineering Corp.
Crane Co.
Crystal Oil Burner Corp.

Davison Gas Burner & Welding Co., N. C.
Domestic Engineering
Dunham Co., C. A.
Struthers Dunn, Inc.

Economy Pumping Machinery Co.
Emerson Electric Mfg. Co.
Engineering Publications, Inc.
Estate Stove Co., The

Fox Furnace Co.
Fuel Oil Journal

Gas-Aire, Inc.
General Electric Co.
General Gas & Light Co.
Grinnell Co., Inc.

Hart Oil Burner Corp.
Heating Journals, Inc.
Heating, Piping & Air Conditioning
Heating & Ventilating
Henry Furnace & Foundry Co., The
Hill Co., E. Vernon
Hoffman Specialty Co., Inc.

Illinois Engineering Co.
Iron Fireman Mfg. Co.
Independent Air Filter Co.

Johnson & Co., S. T.
Johnson Service Co.
Johns-Manville Corp.

Kieley & Mueller, Inc.

Leeds & Northrup Co.
Linde Air Products Co.

McDonnell & Miller
Marsh & Co., Jas. P.
May Oil Burner Corp.
Mercoid Corp.
Minneapolis-Honeywell Regulator Co.
Motor Stoker Corp.
Mouat Vapor Heating Co.
Mueller Furnace Co., L. J.
Multicell Radiator Corp.

Nash Engineering Co.
National Radiator Corp.

Oil Heat
Oxweld Acetylene Corp.
Oil Heating Institute

Parker Appliance Co.
Peerless Unit Ventilation Co.
Penn Electric Switch Co.
Pennsylvania Furnace & Iron Co.
Pierce Butler & Pierce Mfg. Co.
Piqua Electric Mfg. Co.
Powers Regulator Co.
Propellair, Inc.

Raymond Co., F. I.
Robertson Co., H. H.
Rochester Circulator Sales Co.

Sarco Co., Inc.
Silent Automatic Corp.
Silent Glow Oil Burner Corp.
Spencer Heater Co.
Stay-Rite Co., Inc.
Sun Radiator Cover Co., Inc.
Superior Steel Corp.
Surface Combustion Co.
Swartwout Co., The

Taco Heaters, Inc.
Time-O-Stat Controls Co.

Union Carbide Co.
United States Radiator Corp.

Warren Webster & Co.
Warm Air Furnace Fan Co.
Weil-McLain Co.
Whittington, Inc., W. P.

York Heating & Ventilating Corp.
Young Radiator Co.
Youngstown Sheet & Tube Co.

CLEVELAND AUDITORIUM

Program
Semi-Annual Meeting

American Society of Heating and Ventilating Engineers

New Ocean House, Swampscott, Mass., June 22-25, 1931

TECHNICAL

(Daylight Saving Time)

Monday, June 22, 1931

9:30 a. m. Registration.

12:30 p. m. Luncheon (Officers, Council and Authors).

2:00 p. m. Council Meeting.

3:00 p. m. Meeting Committee on Research.

Tuesday, June 23, 1931

8:30 a. m. Registration.

9:30 a. m. Technical Session.

Greeting by D. S. Boyden, Chairman, Committee on Arrangements.

Response by President W. H. Carrier.

Flow of Steam through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger.

Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett.

Sizing Pipes and Orifices for Gravity Hot Water Heating Systems, by E. G. Smith.

Using Gas-Vapor Mixtures for Heating Purposes, by C. A. Dunham.

Report of Committee on Increase of Membership, C. W. Farrar, Chairman.

Report of Committee on Testing and Rating Unit Ventilators, John Howatt, Chairman.

7:00 p. m. Conference of Chapter Representatives.

7:00 p. m. Meeting of Committee on Ventilation Standards.

Wednesday, June 24, 1931

8:00 a. m. Meeting of Nominating Committee.

9:30 a. m. Technical Session.

Essential Elements for Determining Heating Plant Requirements, by F. B. Rowley.

Report of Committee on Rating and Testing Concealed Gravity Type Radiation, R. N. Trane, Chairman.

Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock.

Experiments to Determine Heating Effect Factor, by R. N. Trane and C. J. Scanlan.

Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar.

Thursday, June 25, 1931

9:30 a. m. Technical Session.

Report of Committee on Ventilation Standards, by W. H. Driscoll, Chairman.

Report of Committee on Research, C. V. Haynes, Chairman.

Reports of Technical Advisory Committees.

Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant.

Infiltration through Double Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta.

Measurement of the Flow of Air through Registers and Grilles, by L. E. Davies.

Predetermining the Airation of Industrial Buildings, by W. C. Randall and E. W. Conover.

Unfinished Business.

New Business.

Adjournment.

ENTERTAINMENT

Monday, June 22, 1931

12:30 p. m. Luncheon (Officers, Council and Authors).

8:30 p. m. Informal Reception and Dance.

Tuesday, June 23, 1931

10:00 a. m. Clock Golf.

1:30 p. m. Coach and Boat Trip to Marblehead and Gloucester, including the famous North Shore Drive.

2:00 p. m. Golf—Kickers' Handicap.

8:00 p. m. Bridge Party for Ladies.

Wednesday, June 24, 1931

9:00 a. m. Motor Trip to Historical Boston and Boston Navy Yard to inspect the frigate *Constitution* or *Old Ironsides*.

2:00 p. m. Golf—Research Cup Tournament.

7:00 p. m. Banquet and Dance at New Ocean House (informal).

Thursday, June 25, 1931

9:00 a. m. Special Golf Tournament—Ladies.

2:00 p. m. Trip to historic Lexington, Concord and Salem.

Local Chapter Reports

Cleveland

June 5, 1931. The June meeting of the Chapter was held in the rooms of the Cleveland Engineering Society, and was called to order by Pres. H. F. Morris.

The report of the Auditing Committee was accepted by unanimous vote and three new members, Marion I. Levy, J. E. Maynard, and J. J. La Salvia were elected to membership in the Cleveland Chapter.

The question of sending representatives to the summer meeting at Swampscott, and the method of paying expenses of the members representing the Society involved considerable discussion, and a resolution by C. F. Eveleth to have money appropriated to pay the expenses of one man was amended by T. A. Weager to have the President appoint one representative who would have $25.00 paid toward expenses.

The officers for the year 1931-32 were elected as follows:
President—W. E. Stark.
Vice-President—C. J. Deex.
Secretary—R. G. Davis.
Treasurer—H. M. Nobis.
Board of Governors—F. H. Morris, C. Gottwald and S. H. Givelber.

The speaker of the evening was W. E. Stark, research engineer, Bryant Heater Co., Cleveland, who talked on the subject, What Is Doing in Gas Today.

In discussing the subject, Mr. Stark first presented statistical data showing the striking growth of the gas industry, both manufactured and natural; and particularly the rapid increase in the number of consumers availing themselves of manufactured gas as a house-heating fuel. In picturing the growth of the natural gas industry, he related a number of facts and incidents in the early history of natural gas production, followed by a portrayal of the steadily growing net work of pipe-lines reaching out from the fields of the southwest as far north as Chicago.

A phase of the natural gas business that was touched upon was the liquefied petroleum industry, which renders available and utilizes the valuable by-products of the natural gasoline industry, propane and butane, as the basic ingredient of a gaseous fuel suitable for general distribution. An example is the distribution of butane-air mixtures.

The use of gas in refrigeration and in summer air conditioning for residences was considered and illustrations of modern domestic gas consuming heating appliances were shown.

Michigan

May 13, 1931. The annual outing and election of officers of the Chapter took place at the Hawthorne Golf Club, where 90 members and guests spent a delightful afternoon and evening.

The golf tournament in charge of R. K. Milward and W. G. Boales was unusually successful. Among the members, H. E. Paetz, who scored 90, had the low score, Tom Brown was a close second with 91, and W. C. Salisbury was third with 95. Eddie Anderson received a score of 85, which was the low score for the guests.

The Chapter was particularly fortunate in having as its guest, Dean F. Paul Anderson, Lexington, Ky., past-president of the Society, who paid a very high tribute to the work which the Society, as a whole, is doing and predicted that the future held wonderful things in store for the members of the heating and ventilating and air conditioning profession.

The principal business of the evening was the election of officers. Those elected to serve during the year 1931-32 are as follows:
President—G. H. Giguere.
Vice-President—W. J. Whelan.
Treasurer—E. H. Clark.
Secretary—G. D. Winans.
Board of Governors, N. B. Hubbard, L. L. McConachie and H. E. Paetz.

Following the election of officers, G. H. Giguere addressed the Chapter, thanking the members for the honor they had bestowed upon him and asking for their support during the coming year.

Minnesota

May 19, 1931. The regular monthly meeting of the Chapter was held at the Minneapolis Country Club where the members and guests enjoyed the afternoon and evening together. The afternoon was devoted to golf, which was followed by the annual dinner.

The following officers were elected to serve the Chapter for the coming year:
President—H. E. Gerrish.
Vice-President—W. F. Uhl.
Secretary-Treasurer—A. B. Algren.
Board of Governors—D. M. Forfar and M. E. Bjerken.

Western New York

May 18, 1931. The May meeting of the Chapter, held at Hotel Buffalo, was as usual observed as Kentucky Night. Dean Anderson and several professors accompanied by 60 senior students of University of Kentucky were the guests of the Chapter.

President Davis very ably acted as toastmaster and introduced Mr. Rathman, Commissioner of Public Works, who acted as personal representative of Mayor Roesh who was unable to be present. Mr. Rathman expressed the regrets of the mayor and personally extended the official welcome of the City of Buffalo to Dean Anderson and his party.

President Davis then introduced Sam Botsford, vice-president of the Buffalo Chamber of Commerce, who gave some interesting opinions regarding the relation of the engineer to the business depression.

The toastmaster then introduced Dean Anderson, who spoke briefly, expressing his gratitude to the Chapter for again having his seniors as their guests. The Dean replied to Mr. Botsford's statement, that the unemployment situation was due in a large measure to engineers, who have developed mechanical labor saving devices to their present high efficiency, saying, that the industry asked for efficient mechanical devices and the engineers fulfilled the request so completely that the business man is unable to keep pace with engineering progress.

Colonel Morrow, a fellow Kentuckian, gave an interesting talk on the friendship between Kentuckians and the Niagara Frontier, dating back to Civil War Days. Colonel Morrow is a very forceful speaker and his address resulted in a hearty response from the audience.

C. W. Farrar, chairman of the Membership Committee, and Roswell Farnham, member of the Council, were introduced, each of whom responded briefly.

L. A. Harding, past president of the Society, was presented as the speaker of the evening. His subject was the Utilization of the Sun's Energy. He explained some of the developments which have been going forward for many years to harness the sun's energy. Due to the lateness of the hour it was impossible for Mr. Harding to present his paper in full, which the majority of the members and guests regretted.

The attendance at this meeting was unusual, there being 148

members and guests present, which was the largest in the history of the Chapter.

Philadelphia

May 14, 1931. The regular meeting of the Chapter was held in the form of the annual outing, and took place at the Tavistock Country Club, Haddonfield, N. J.

The weather was far from being favorable, but approximately 25 members braved the weather to participate in the golf tournament. J. H. Hucker, J. Walsh, George Lewis and William Ricard were among those who finished with low scores.

A dinner was served at the Club, after which the Chapter adjourned to the plant of Warren Webster & Co., where an inspection was made. At the conclusion of the inspection the members and guests assembled in the Exhibition Room where H. F. Marshall, Warren Webster Co., presented an illustrated talk on *Steam Consumption of Heating Systems.* Mr. Marshall's talk was thoroughly enjoyed by all present.

Pittsburgh

May 11, 1931. Thirty-two members and guests attended the May meeting of the Chapter, which was held at the Fort Pitt Hotel.

President Houghten, who presided over the meeting, called attention to the return of E. C. Evans, who left Pittsburgh a year ago to go to California, but who has returned to be with the Pittsburgh Chapter.

Following several announcements regarding the Semi-Annual Meeting of the Society, Swampscott, Mass., F. A. Gunther, chairman of the Program Committee, suggested that although the next regular meeting of the Chapter would be held in September, it was planned that a meeting be held during the summer for the members, their wives and guests, with either golf or a dinner-dance for entertainment.

Mr. Gunther then introduced Prof. S. E. Dibble, Carnegie Institute of Technology, and past-president of the Society, who addressed the meeting on the subject of *Concealed Radiation—Types, Construction and Capacities.* Professor Dibble stated that in the design of concealed radiation one must strive to get the proper relation between the steam or primary surface and the extended surface, and that an important point in the design is the impingement of the air to obtain a lower surface resistance so as to increase the output of the radiator. Professor Dibble also stated that fusion or metallic contacts between the prime surface and the extended surface had proved by test to be much better than frictional contacts, or contacts obtained with either solder or paint.

After many interesting questions were raised and discussions given, a rising vote of thanks was tendered Professor Dibble by the members and guests in appreciation of his interesting talk.

St. Louis

May 15, 1931. Twenty-eight members and guests assembled at the Roosevelt Hotel for the May meeting of the Chapter.

In the absence of President Rosebrough, James Foster called the meeting to order, and after reading the minutes of the previous meeting, which were approved as read, L. Walter Moon, chairman of the Program Committee, suggested a trip to Port St. Louis, to inspect the Airplane Hangars and have dinner at Skyway Inn, which would be followed by bridge or dancing. This would be held instead of the usual Ladies night for the June meeting. A motion to this effect was made and carried.

Several communications were read, among which was one from Secretary Hutchinson, requesting the President and Secretary of the Chapter to attend the Semi-Annual Meeting in Swampscott, June 22 to 25, at which time there would be a meeting of the Chapter Relations Committee.

Following the Report of the Membership Committee given by George Myers, the meeting was turned over to L. Walter Moon, who introduced W. Rodney Smith, chief engineer, Wm. B. Ittner, Inc., who read a paper dealing principally with the evolution of school heating and ventilating in the St. Louis schools, with which he was connected for 17 years. Mr. Smith described the original cannon ball stoves used over 30 years ago and the different types used since that time and ended with a description of present standards and stated, that the St. Louis schools today have the best school system in the world, due to the elimination of bonded indebtedness. In conclusion Mr. Smith answered numerous questions presented by the members and guests, which brought about an interesting discussion.

It was moved by C. A. Helwig and seconded by Mr. Carlson, to request Mr. Smith to prepare this paper for the Annual Meeting of the Society in January, and present it to the Council and Secretary for their consideration. The motion was carried, and a vote of thanks was extended to Mr. Smith for his able handling of this subject.

Death of Wallace M. Cosgrove

It is with profound sorrow that the Society learns of the death of Wallace M. Cosgrove, vice-president of the American Radiator Co., who passed away June 11, 1931, at the Orange Memorial Hospital, where he underwent a serious operation.

Mr. Cosgrove was born in Haverstraw, N. Y., in 1873, and received his early education in the public schools of Jersey City. He spent his entire life in the heating industry and from 1888 to 1907 he was connected with A. A. Griffin Co. During the year 1907 he became associated with the New York Branch of the American Radiator Co., of which he was manager for a good many years and one of the most widely known men in the trade.

He had been a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS since 1922, and one whom the Society will always remember, for he was well liked among his fellow members.

Besides being a member of this Society, Mr. Cosgrove belonged to the Bergen Lodge, F. and A. M., the New York Rotary Club, the Engineers' Club of New York and the Cartaret Club of Jersey City. He was also a member of the board of governors of the Canoe Brook Country Club, a director of the Mount Tabor Field Club, and of the *Plumbing and Heating Golf Association.*

The funeral services for Mr. Cosgrove were held on Sunday, June 14, 1931, at the South Orange Methodist Episcopal Church. The pastor, Rev. George G. Vogel, assisted by the Rev. Dr. Harry Everett, of Jersey City, officiated at the services.

Mr. Cosgrove is survived by his widow and two sons, John P. and Kenneth M. Cosgrove, to whom the Council and Officers of the Society extend their sincere sympathy.

E. P. Bailey to Manage Gas Division

Commencing on June 1, Edward P. Bailey, Jr., started his duties as General Manager of the Gas Heating Division of National Radiator Corp., Johnstown, Pa., according to an announcement recently made by the company.

Mr. Bailey brings a wide experience in the gas field to his new position through his association with Bryant Heater and Manufacturing Co.

Langenberg Heads Engineering Company

A new company has been organized by E. B. Langenberg in St. Louis, the E-B Engineering Co. with offices at 6625 Waterman Ave., St. Louis, and will supply designs and specifications to manufacturers, dealers and architects, particularly for warm air heating installations.

The many friends of Mr. Langenberg will wish him the greatest success in his new venture.

Of Interest to Members

Thornton to Manage Westinghouse Association Activities

Frank Thornton, Jr., manager of residence engineering for the Westinghouse Electric and Mfg. Co., Pittsburgh, Pa., has been appointed manager of association activities of that company.

Mr. Thornton was graduated from the University of Missouri in 1908 with the degree of B. S. in electrical engineering. He entered the employ of the Westinghouse Electric and Mfg. Co., and, after completing the graduate student course, was assigned to the engineering department, where he specialized in electric heating. When the electrical heating engineering department was organized, he was appointed manager of it.

Mr. Thornton is a fellow of the *American Institute of Electrical Engineers* and a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the Edgewood Country Club, and the University Club, Pittsburgh.

Fire Underwriters Issue New Building Code

The fifth edition of the building code recommended by the National Board of Fire Underwriters, New York, has recently been issued. This ordinance provides for fire limits and regulations governing the construction, alteration, equipment, or removal of buildings or structures.

Mechanical ventilation is covered in Section 505 of Article V, Light and Ventilation, as follows:

(a) When mechanical ventilation is required, or is permitted as an alternative, the system shall be designed and constructed in accordance with generally accepted good practice, to provide the necessary changes of air, but not less than prescribed in this section.

(b) Except as otherwise specifically prescribed by law or in duly promulgated rules, the recommendations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS shall be deemed to be generally accepted good practice.

The code also includes information on chimneys and flues and heat appliances and forced draft systems.

New Edition of A O B A Handbook

The *American Oil Burner Association* has recently issued a revised edition of its Handbook of Oil Burning by H. F. Tapp, the purpose of which is to present facts and authoritative information pertaining to oil burning.

Considerable data derived from the A. S. H. V. E. GUIDE have been included. The contents of this new edition of the Handbook of Oil Burning are as follows:

Oil as a Fuel; Sources, Characteristics and Properties of Oil Fuel; Characteristics and Properties of Solid, Gaseous and Other Fuels; Comparative Fuel Costs; Chemistry of Combustion and Flame; Fundamentals of Heat and Heat Transfer; The Determination of Heating Capacity Requirements; Estimating Fuel Requirements for the Heating Season; Heating Systems; Boilers and Warm Air Furnaces; Drafts and Chimneys; Industrial Oil Burner Applications; Oil Burner Types and Principles of Construction; Oil Burner Controls and Motors; For the Architect; Ordinances and Regulations on Oil Burners and Equipment; Fuel Oil Tanks and Storage; Pumps, Preheaters, and Piping; Preliminary Surveys and Estimates; Installation Methods and Refractories; Tests and Test Codes; Sales and Service; Accounting and Business Law; Miscellaneous Information and Tables and Bibliography.

Much credit is due those responsible for this handbook for the useful manner in which the data are compiled.

A E C Aiding Engineering Students

A communication was recently addressed by the *American Engineering Council* to the 111 known engineering and allied technical societies in the United States, asking what, if anything, they were doing to aid worthy young men to obtain a technical education. Replies were received from 77 organizations. Of this number, 14 were providing scholarships in some form, and 16 were doing one or more of the following: offering loan funds; night courses; prizes; a special series of lectures for students in their respective localities.

In many cases the information available concerning scholarships, prizes and loan funds for engineering students was not sufficiently complete to describe in detail the work of various engineering societies and clubs in this worthy field.

Standards of Hydraulic Society Issued

The sixth edition of standards recommended by the *Hydraulic Society* is announced.

In accordance with its aim to be of service to engineers, buyers and users of pumps, as well as to its own members, the *Hydraulic Society* has collected pertinent technical, engineering and commercial pump data; it recommends standard definitions, terms and practices where such action appears to be feasible; and it has compiled this information in a *Standards* pamphlet for distribution to those interested.

The first edition was a nineteen page booklet, 6 in. x 9 in., issued in 1921. As the usefulness of the *Standards* has been demonstrated, additional information has been included in succeeding issues, culminating in this sixth edition of 96 pages in 8½ in. x. 11 in. size and in which, through the courtesy and cooperation of the members of the *Hydraulic Society,* some nonmember pump manufacturers, the *American Society of Mechanical Engineers* and other sources, an effort has been made to produce a publication that will merit the approbation of the engineering profession.

Commercial as well as technical material in the pamphlet is submitted as being the opinions and recommendations of the *Hydraulic Society* as to what is considered good practice in the pump industry, it being recognized that the *Hydraulic Society* cannot and does not make or enforce any agreements that might be considered in restraint of trade or in violation of the antitrust laws.

It is believed, however, that a general adoption of the recommendations in these Standards will be of mutual advantage to the purchasers and users of pumps, and the engineering profession, as well as to the pump manufacturers, and it is hoped that these recommendations will be considered as the standard trade definitions and customs in the pump industry, and will be so used by both buyers and sellers as governing transactions where nothing to the contrary has been established.

Electrical Engineers Elect Officers

C. E. Skinner, assistant director of engineering, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., was elected president of the *A. I. E. E.*, as announced at the annual meeting held at Asheville, N. C., June 22, 1931. The other officers elected are: Vice-Presidents, W. B. Kouwenhoven, Baltimore, Md.; W. E. Freeman, Lexington, Ky.; P. H. Patton, Omaha, Neb.; A. W. Copley, San Francisco, Calif.; L. B. Chubbuck, Hamilton, Ont. Directors, L. W. Chubb, East Pittsburgh, Pa.; B. D. Hull, Dallas, Texas; H. R. Woodrow, Brooklyn, N. Y., and National Treasurer W. I. Slichter, New York City, who was reelected.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 14 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by July 15, 1931, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Cooper, John Raymond, Eastern Branch Mgr., Thermal Units Mfg. Co., New York, N. Y.	R. E. Keyes R. T. Coe	P. J. Bryant A. S. Lau
Gassler, John H., Mgr., Engrg. Dept., Crane Co., Jacksonville, Fla. (*Advancement*)	C. P. Lichty B. A. Thomas	W. L. Hodeaux
Griffin, John J., President, International Engrg. & Supply Co., Inc., St. Louis, Mo. (*Reinstatement*)	G. W. F. Myers V. M. Lathers	Paul Sodemann J. M. Foster
Girrbach, George Frederic, McQuay Radiator Corp., Minneapolis, Minn.	E. F. Jones W. N. Parks	C. E. Gausman A. L. Sanford
Hvosleff, Fredrik Waldemar, Kohler Co., Kohler, Wis. (*Reinstatement*)	G. L. Larson F. B. Rowley	J. F. McIntire R. F. Connell
Kinney, William Hamilton, Asst. Engr. of Subway Design, Bureau of Subways, Board of Local Improvements, Chicago, Illinois	B. L. Casey C. W. Johnson	J. F. Hale L. L. Narowetz, Jr.
Lewis, Will, Munroe Furnace Supply Company, Omaha, Nebr.	I. C. Bostain W. J. Doyle	G. A. Voorhees W. Gunton—*Non-Member*
Morris, Edward Jerome, Student, Carnegie Inst. Tech., Pittsburgh, Penna.	S. E. Dibble F. C. Houghten	Joe Blackshaw S. H. Harper
Naylor, Charles Luzier, Supt., Heating, Light and Power Depts., Atlantic Refining Co., Philadelphia, Penna.	J. C. Hurley M. F. Blankin	H. G. Black A. J. Nesbitt
Oates, Walter A., House Heating Dept., Lynn Gas & Electric Co., Lynn, Mass.	Col. J. B. Coleman J. C. Hartwell	A. W. Moulder C. R. Swaney
Schleyer, Emil Francis, Estimator and Supt., E. W. Mandeville, Inc., Brooklyn, N. Y.	F. W. Phillips, Jr. L. W. Atwater	B. H. Griffen H. W. Maxfield—*A. S. M. E.*
Shimanski, Victor Edward, Sales Engr., Trane Co., La Crosse, Wis.	N. D. Adams R. N. Trane	T. R. Johnson M. W. Miller
Van Alsburg, Jerold Henry, Hart & Cooley Mfg. Co., Holland, Mich.	V. W. Cherven F. R. Bishop	Alfred Knowles—*Non-Member* H. S. Covell—*Non-Member*
Waters, George G., District Manager, American Blower Corp., Pittsburgh, Penna.	W. A. Rowe	

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

Armstrong, Asher D., Mgr., Heating Department, Crane Co., Minneapolis, Minn.

Ashley, Carlyle Martin, Engineer, Carrier Engrg. Corp., Newark, N. J.

Blitz, Emmanuel, Engineer, Andrew J. Thomas, Archt., New York, N. Y.

Cunningham, Tom M., Branch Mgr., Carrier Engrg. Corp., Dallas, Tex.

Keefe, Edmund T., President, Underground Steam Construction Co., Boston, Mass.

Maynard, J. Earle, Heating Engr., Fox Furnace Co., Elyria, Ohio.

ASSOCIATES

Brown, Thomas, Contracting Engr., Own Business, Toronto, Ont., Canada.

Burton, W. Dean, Chief Engr., Combustioneer, Inc., Goshen, Ind.

Giffen, J. Kerr, Stanton Heater Co., Martins Ferry, Ohio.

Kelly, Joseph A., Asst. Chief Engineer, John J. Nesbitt, Inc., Philadelphia, Penna.

Myers, Eddie Vern, Vice-Pres. & Supt., A. S. Hunter & Son, Inc., Willard, Ohio.

Rhea, Chester A., Heggie-Simplex Boiler Co., Philadelphia, Penna.

JUNIORS

Edwards, Don Calvin, Jr., Sales Engr., Carrier-York Corp., Nashville, Tenn.

Levy, Marion I., Bishop & Babcock Sales Co., Cleveland, Ohio.

McElgin, John Wilbert, Asst. to Experimental Engineer, John J. Nesbitt, Inc., Philadelphia, Penna.

Powers, Edgar Chester, Sales Engr., Rome Radiation Co., Philadelphia, Penna.

Heating · Piping and Air Conditioning EDITORIALS

The Future of Refrigeration

"In my most vivid recollections of my Mother I see her going down a steep hill to a cool spring three times a day to get the butter and milk for our meals." That from a biography of a noted man brings back to us the conditions that existed not many years ago and which indeed still exist in many parts of the country.

The refrigeration industry has grown in the meantime to eleventh place among our national industries. It has revolutionized the food industry. As a factor in air conditioning it has entered industry, the theater, the office building, the hotel and the home and has provided summer comforts that heretofore were not available to kings and queens.

And yet it is stated by men prominent in the refrigeration industry that refrigeration has just made a start and that there will be developments which will make our former refrigeration progress insignificant by comparison.

Even though one may not be a prophet he can reasonably foresee—if not predict—certain obvious developments.

More people every year are demanding summer cooling for occupied spaces. How soon will summer comfort be made available to practically everyone? What will this mean in terms of greater efficiency, better and more work with less effort than in the enervating heat of summer? What will it mean in increasing the mental and physical activity of tropical races and tropical industries that are trying to survive? What will it mean in the development of the great resources of tropical regions?

One might also speculate on the possibilities from a pathological point of view. Many diseases of the human family are accelerated by excessive pressures, temperatures, humidities, air velocities or dust in the air. So it is reasonable to suppose that many diseases may be retarded or perhaps cured by accurate control of these factors.

Truly, the field of refrigeration and air conditioning merely has been scratched. We are constantly discovering new uses and new benefits.

For centuries we have known the mechanical equipment to use for cold weather comfort and even the cave man knew how to make clothing to protect his body against the chill of winter. But not until the twentieth century did we learn the mechanical means for defending ourselves against excessive climatic heat and we have not yet learned, (and probably never will) how to make a body covering for adequate heat protection. Until recently a Northerner was more defenceless against the scorching sun of the tropics than were the Esquimaux a thousand years ago against cold.

We look back thirty centuries to the birth of heating for human comfort. Probably human beings three thousand years from now will celebrate this century of outstanding engineering achievement. They may honor it for the development of a science with human benefits more profound than heating has ever had—the science of refrigeration.

Careful Instruction

Frequently, the heating, piping or air conditioning contractor must spend time and money investigating an installation when the fault can be traced to improper operation. Contractors deserve the hearty cooperation of owners in making sure that operating staffs are carefully instructed in the proper methods of caring for these important divisions of the mechanical equipment of buildings.

Proper operation demands not only technical skill, but extensive knowledge of the individual peculiarities of each installation. Its exercise means reducing costs for the owner, reducing contractors' expenditures and leading to economies and advantages to all concerned.

Comfort on Wheels

A progressive—though inevitable—step is described in a pictorial feature elsewhere in this issue. We refer to the air conditioning of a complete New York to Washington train. While a few leading roads have conditioned—or are considering conditioning—dining cars this is the first train to offer comfortable atmosphere from head to rear. It is being brought to the attention of the public through extensive advertising.

We hope that other railroads will be quick to follow the lead. And if anything can be done about suburban coaches, too, we'll offer a prayer of thanks to the air conditioning engineer at 5:15 p.m. daily (except Sundays).

Practical Piping Problems

Undetected Leaks Waste Money

W. H. Wilson's Monthly Discussion of the Problems He Meets From Day-to-Day

Leaks in piping installations that ordinarily are not visible are a major cause of operating losses and a source of piping trouble. Leaks that are visible are generally sure to receive immediate attention and prompt repairs. This cannot be said of the leaks that are not readily detected, as they may be in tunnels, pits, basements and places not readily accessible. Noise from equipment and machinery in operation frequently renders it difficult to detect steam, air or water leaking through pass by valves, cross connection and division valves, and they may pass unnoticed for long periods of time.

Illustrations Show Results of Leaks

The accompanying illustrations show valve seats and discs that have been worn away by steam and water leakage. This calls attention to the fact that proper plant operation, maintenance inspections and prompt repairs were lacking.

Fig. 1 shows a seat and disc from a 1-in. globe valve that was used as a pass by valve on a steam line, 150 lb. working steam pressure. The metal valve seat is slightly worn, and the disc is worn badly; much steam pressure was lost through this valve.

The leak was not detected for some time, as the valve was located overhead, near machinery in continuous operation. Also, this connection was connected to a discharge line used to receive the discharge from several traps and consequently the leaking valve was not noticed for some time. This seems to question the advisability of installing a pass by valve and connection for a steam trap.

Fig. 2 shows a metallic seat and disc that was removed from a small globe valve, used on the steam supply to a pipe coil submerged in a tank for heating a solution used in connection with nickel plating equipment. This valve was kept closed after regular working hours, but the leakage through the seat and disc kept the solution in the tank at a high temperature, continuously, nights and over the week ends, with a consequent loss of the solution by evaporation, as well as a

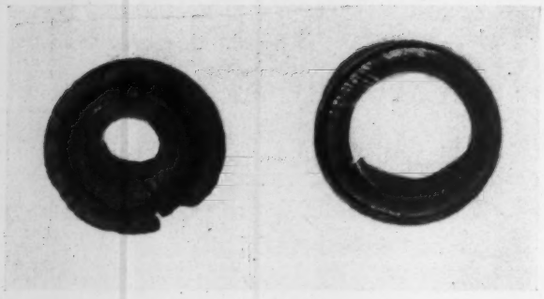


FIG. 1—SEAT AND DISC FROM A VALVE ON A STEAM LINE; THE LEAK WAS NOT DETECTED FOR SOME TIME

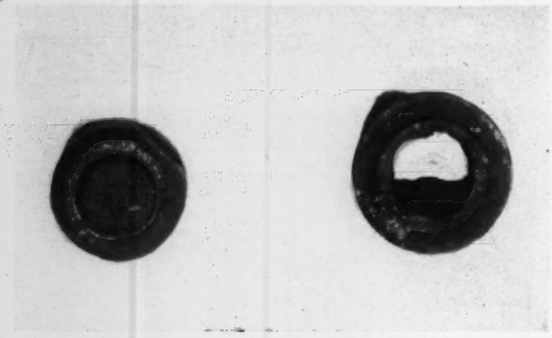


FIG. 2—SEAT AND DISC REMOVED FROM VALVE ON STEAM SUPPLY TO PIPE COIL SUBMERGED IN A TANK FOR HEATING A SOLUTION. SOLUTION WAS LOST BY EVAPORATION AS WELL AS STEAM BEING LOST

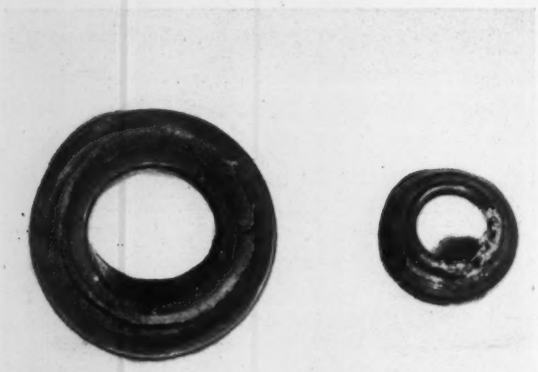


FIG. 3 (LEFT)—IT TOOK ONLY A SHORT TIME TO REPLACE THIS WORN REDUCING VALVE DISC. FIG. 4 (RIGHT)—MANY GALLONS OF WATER WERE WASTED BEFORE THIS SEAT FOR A HOT WATER FAUCET WAS REPLACED

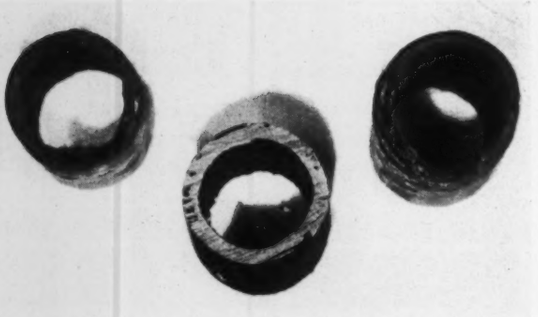


FIG. 5—NIPPLE WORN BY STEAM AND CONDENSATION

corresponding loss of steam, the main steam supply line being kept on for other purposes. No leakage of steam being visible, as the heating coil was provided with a steam trap discharging into a return line, it was taken for granted that the heating equipment was in order.

Fig. 3 shows a metallic disc removed from a steam reducing valve. On account of the bad condition of this disc, the steam pressure would rise on the delivery side of the reducing valve when a small amount of steam was required. For some time this was being taken care of by throttling the shut-off or stop valve controlling the steam supply to this reducing valve. There was no reasonable excuse for this practice, as it took only a short time to replace the worn disc in the reducing valve with a new one.

The small valve seat shown in Fig. 4 is a metal seat removed from a hot water faucet on a wash basin in a lavatory. This equipment was not regularly inspected by the maintenance employes and the faucet was allowed to remain in this condition for some time. It was estimated that 8000 gallons of hot water per month was wasted in this manner.

Fig. 5 shows a wrought pipe nipple, of extra strong pipe, that was removed from a submerged steam pipe coil in a tank used for heating water. The steam and condensation from the steam passing through the pipes has worn both ends of the nipple, so that it is cut through into the interior of the pipe, at the threaded ends, causing some of the steam and condensation to pass into the water in the tank. The portion of the pipe in the nipple between the threads is in good condition.

This tank was provided with an overflow pipe and the surplus water passed out through this overflow and the steam leaking in the pipe coil was not discovered for some time.

Supervisors in charge of plant maintenance are usually busy in computing the cost of operation of steam generating equipment, the operation of prime movers and items of large magnitude, yet it pays to devote attention to leaks. When they remain unattended for long periods of time, they are costly.

Pipe Welding Booklet Published

A 44-page booklet "Pipe Welding" has been published recently by the Gas Products Association, 250 E. Ontario St., Chicago. The manual contains, in condensed form, information on the various phases of welding steel and wrought iron piping systems for industrial use. Tables of cost data for welding and cutting pipes from ⅜-in. to 30-in. diameters are included.

Among the topics are types of joints; bevelling pipe; expansion and alignment; spacing; clamps; tacking; templates; fabrication of headers; connecting branch to header; swaged reducers; closing end of line; special fittings for welding; welding flanges; expansion bends and anchors; designing a welded lay-out; qualification tests; pipe welding costs for refrigeration, power, heating, industrial and long transmission piping; and cutting cost data for standard weight and extra heavy pipe.

The booklet is paper-covered and is 4¼ by 6½ in. Its price is fifty cents.

Process Industries Piping

By C. L. Mantell*

PROCESS industries are generally thought of as being those in which chemical processes and reactions play a major part. Their diversification is wide; from those strictly chemical concerning themselves with the manufacture of acids, bases, salts, organic products, pharmaceuticals, and fertilizers; through those which are electrochemical, dealing with the refining of metals, the production of chlorine and caustic, and the electrolysis of fused salts from which aluminum, magnesium, and sodium result; those which are electrothermal in nature, such as the manufacture of ferroalloys, calcium carbide, and cyanamid; to others somewhat less predominately chemical, in which petroleum refining, sugar preparation, the manufacture of pulp and paper, paints, inks, pigments, leather, cement, the treatment of water and disposal of sewage, commercial fermentations, and a host of others are included. The entire group is thought of as the chemical engineering process industries.

Substances Piped

In fields other than chemical engineering, pipe lines carry substances such as air, water, or steam, having little or no destructive character. As a result, a small number of materials serve satisfactorily. In the case of chemical processes, however, a very large and varied number of materials must be transported through pipes. These include the mineral acids, such as sulphuric, hydrochloric, and nitric, in various concentrations; organic acids, such as acetic, lactic, tartaric, butyric, and oxalic; corrosive gases, such as chlorine, bromine, sulphur dioxide; solutions of salts which attack metals, such as brines; as well as a host of other materials among which are caustic, bleaching powder solutions, hypochlorites, and miscellaneous products.

Wide Variety of Piping Materials Used

The transportation of these substances, in both concentrated and dilute forms, commercially pure, slightly or badly contaminated with other substances having corrosive action, necessitates unusual piping materials and layouts. It is the common thing to find in chemical process plants piping made of special alloys, lead, bronze, tin, aluminum, nickel, chromium and other alloy steels, ceramics, plastics, glass, rubber, silica, and many other materials.

Factors to Consider

The piping engineer is concerned very largely with the physical properties of piping, its strength, elasticity, expansion and contraction, and its resistance to corrosion. The process plant engineer must consider (in addition to the physical factors) the corrosion of the material particularly, its ability to withstand chemical attack over a wide range of conditions, among which are concentration of the substance being handled, temperature, pressure, humidity, alloy composition, and structure. The use of unusual products for piping brings in its train problems of joining, connecting, and fitting together of lines.

The process piping engineer is intensely interested in corrosion, its causes and effects, in the belief that this knowledge will enable him to discover, develop, and use better materials of construction for transportation of substances through pipes. The present day literature on corrosion is vast, but it is not readily useful because it is poorly coordinated. Most piping materials in use have been employed as the result of costly experimentation and plant experience. Few metals or alloys are adaptable to a wide range of different chemicals and conditions. Many of them are very serviceable for specific purposes. Their physical properties necessitate design details differing to a considerable degree from those found in steam, air, and water pipe lines.

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Some Odd Uses for Lead Pipe

To obviate the necessity of stocking a supply of spare gaskets of odd shapes, a coil of small size, medium weight lead pipe may often be handily used; if lack of space prevents the use of a gasket of this thickness, the pipe may be split in two parts and hammered into sheet lead which will usually suffice until a regular gasket is obtained.

Asbestos rope packing inside a piece of lead pipe provides a suitable gasket for the tongue and groove or similar flange joints. The difficulties usually encountered in the removal of an old gasket in this type of joint are lacking when lead pipe and asbestos rope are used; when a gasket of this type is replaced, if any of the old lead remains in the groove and due to the lack of spring to the pipe line it is difficult to properly clean the joint, the application of a blow torch will soon have any lead deposits running out of the bottom of the groove.

The small pieces, often left over from jobs of lead piping, ordinarily considered as scrap and remelted, may be filed and used to good advantage when combined with oil. Leaks through the threads of stud bolts may be prevented and their easy removal permitted by dipping the bolt in oil and then in a small can of lead filings before inserting; this method is valuable in case of loose threads, as the lead filings stick to the oil and the lead provides a filler to prevent leaks and rusting of the threads.

Lead filings and oil brushed on a gasket permit its removal intact at any time, except in lines where excessive heat is employed. The gasket is lightly brushed with oil and a small amount of lead filings smoothed evenly over the oil.

In the absence of good dies for pipe, leaky threads may be produced. Lead filings and oil may be used to help these poor threads to make a tight joint. After the pipe is threaded, it is usually coated with a film of oil and dipped into a small can of lead filings. The lead filings stick to the oil and the heat of the joint is sufficient to soften the lead to provide a filler. These lead filings are an excellent means of making brass pipe threads tight. —*William Anderson.*